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NBS/ESDS - A Computer Model for the Simulation of Hot Water Systems

U.S. DEPARTMENT OF COMMERCE
National Bureau of Standards
National Engineering Laboratory
Center for Consumer Product Technology
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ABSTRACT

A computer model, entitled NBS/ESDS, has been developed using a finite difference method to simulate the generation, storage and distribution of a hot water in a system commonly found in many residences, and some commercial facilities.

The conduction heat equation which governs the heat transfer in the water heater during all modes of operation, and the heat transfer in the distribution pipe network during its cooldown periods, is solved. A solution is also obtained for the pipe flow equation which governs the heat transfer in the distribution pipe network during the draw periods. Thermal stratification of water is modeled, and slug flow is used to simulate the draw of heated water from the water heater.

A parametric study is also conducted on a given hot water system for a given draw schedule during a 24-hour period. The effects studied include insulating the hot water heater, insulating the distribution pipe, and incorporating a heat trap. Effects of concentrating the distributed draws to two short periods of time of the day on the hot water delivery and the heat loss with the same energy consumption are also studied.

I. Introduction

Domestic water heating is the second largest energy use in the home—ranked only after space heating. It accounts for about 15% of the total residential energy consumption and 3% of the total U.S. energy use (1). In addition, water-heating accounts for about 8% of commercial energy consumption. In total, water heating accounts for more than 4% of the U.S. energy consumption. The energy cost for water heating, however, is often over-shadowed by the costs of space heating, air conditioning and lighting—it is hidden in the utility bills covering the more significant energy usage.

Of the annual energy dollars spent for water heating in the domestic, storage-type water heater, a significant portion is lost during standby just to maintain the hot water ready for use. The heat lost during the standby phase constitutes more than 15% of the total energy for the electric water heater and more than 25% for the gas water heater (2).

Another significant loss is due to the distribution pipes. Their heat loss occurs during both the draw and the non-draw cycles in a typically daily hot water use cycle. During a draw cycle, hot water enters and flows along the distribution pipes from the hot water heater. As a segment of hot water travels inside the pipe it loses heat both to the pipe walls and through the pipe walls to the ambient. Thus, when this segment of water reaches its draw points, its temperature is much lower than when it leaves the water heater. Immediately after a draw cycle terminates, the average temperature of water in the pipe is between the water temperature in the water heater and the water temperature at the draw point--usually much higher than the ambient. A good portion of the thermal energy of the water in the pipe is lost to the ambient if this non-draw cycle lasts more than a few minutes. A few quantitative questions need to be addressed on the pipe heat transfer:

1. How much energy is lost during a draw cycle?

2. How much energy is lost during a non-draw (or cooldown) cycle?

3. How effective is the pipe insulation in reducing heat loss during the draw and non-draw cycles?

4. How may the hot water delivery be affected by the different draw schedules?

5. How does the configuration of the distribution pipe (including the heat trap) may affect the heat loss?

The loss from one system in the house may be considered as a gain for another system. For example, the heat loss from the hot water system may be considered as a heat gain to the living quarters during a heating season. During a cooling season, on the other hand, the same heat loss is not only a loss of the hot water system but also an added load to the cooling system of the living quarters. To address the question of what constitutes heat loss the entire house (including all the subsystems) should be studied as one integrated system. Only by such means could the interaction among all the subsystems may accurately determined. While the present model of the hot water system does not consider the cooling and heating system of the house as a part of the computing iteration loop, it does compute accurately the amount and distribution of heat loss to the house. This heat loss could then be used as an input to a model of the cooling and heating system of the house.

II. Bulk Behavior of the Water Heaters

A basic understanding of water heater standby and recovery may be obtained by considering the bulk behavior of the water heater as given in this section. This approach has many limitations. A more detailed study of the water heater system which considers the dynamics of draw, temperature dependent conductance and variable heat capacitance of the water heater composite wall along with a parametric study of the tank structure and distribution system are given in the following sections.

Suppose only the bulk behavior of the water heaters is to be studied, then the water in the storage tank may be considered as at one uniform temperature, T. If the energy in the water stored above room temperature is Q and the heat transfer rate through the walls, is q, then during cooldown

$$\frac{dQ}{d\theta} = -q = UA(T-T_a) \qquad (1)$$

where U is the overall heat transfer coefficient, A its associated area, θ time and T_a the ambient temperature. The product UA is sometimes called overall conductance and may be expressed as

$$UA = (UA)_t + (UA)_b + (UA)_o \qquad (2)$$

where the subscripts t, b and o denote top, bottom and lateral surfaces, respectively. A typical overall lateral conductance $(UA)_o$ may be expressed as:

$$(UA)_o = \frac{1}{R_1 + R_2 + R_3 + R_4 + R_5} \qquad (3)$$

where the R's are the individual thermal resistances of the composite layers of the water heater. Namely, R_1 is the resistance between the water and tank walls, R_2 the resistance of the tank walls, R_3 the insulation, R_4 exterior shell and R_5 the resistance between the exterior shell and ambient air. The internal energy, Q, of the water relative to the ambient temperature is expressed as

$$Q = \rho c V (T-T_a) \qquad (4)$$

where ρ is the water density, c the water specific heat and v the volume of the storage tank. Equation (1) is rewritten using equation (4) as

$$\frac{dQ}{d\theta} = -\frac{UA}{\rho cV} \quad Q = -\eta_s Q, \text{ or} \qquad (5)$$

$$\frac{dQ}{d\theta} + {}_sQ = 0 \qquad (6)$$

${}_s$ is the cooldown rate, and is defined as a fraction of the instantaneous stored energy lost per hour. This cooldown rate is different from the standby loss. The standby loss is a measure of average heat loss of the water heater and includes the efficiency of the water heater's heat source to replenish the heat lost. This η_s takes into account only energy lost during the cooldown phase. If the resistances in equation 2 are constants, η_s is a constant, and equation (6) can then be integrated as

$$Q = Q_0 \; e^{-\eta_s \theta} = Q_o \; e^{\frac{-\theta}{\theta_c}}$$

where Q_o is the internal energy content of water at the beginning of the cooldown period, and θ_c is the time constant equal to $1/\eta_s$.

The heat transfer between the water and the tank walls and between the exterior shell and the ambient are functions of temperature. The former is mainly natural convection and the latter is a combination of natural convection and radiation. Therefore, experimental cooldown curves of water heaters do not follow an exponential curve precisely.

A more general form of Equation (6) is

$$\frac{dQ}{d\theta} + \eta_s Q = q_e, \qquad (7)$$

where q_e is a constant strength heat source (or sink). Equation (6), then, is a special case of equation (7) in which the heat source has zero strength. Equation (7) has the following solution,

$$Q = Ae^{-\eta_s \theta} + \frac{q_e}{\eta_s}, \qquad (8)$$

where A is an arbitrary constant. Suppose Equation (7) is subject to the initial conditions of

$$Q = Q_1 \text{ at } \theta = 0,$$

then $A = A_1 - \frac{q_e}{\eta_s}$, thus

$$Q = Q_1 e^{-\eta_s \Theta} + \frac{q_e}{\eta_s}(1 - e^{-\eta_s \Theta}), \text{ or} \tag{9}$$

$$\frac{Q}{Q_o} = \frac{Q_1}{Q_o} e^{-\eta_s \Theta} + \frac{1}{\eta_s} \frac{q_e}{Q_0}(1 - e^{-\eta_s \Theta}) \tag{10}$$

With typical values of:

$$Q_1/Q_o = 0.90$$

$$\eta_s = 0.01/\text{hour}$$

$$\frac{q_e}{Q_o} = 0.41/\text{hour}$$

Then the Q/Qo may be tabulated with respect to Θ (Figure 1).

Θ (hours)	0.0	0.1	0.2	0.2497	0.5	0.75	1.0
Q/Q_o	0.9	0.94	0.98	1.0000	1.10	1.1996	1.299

Suppose the heater is turned off at $\Theta = 0.2497$ hour, then the governing equation (10) reverts to

$$Q/Q_o = e^{-\eta_s \Theta'} \tag{11}$$

where Θ' is time in hours counting from $\Theta = 0.2497$ hour. With $\eta_s = 0.01/\text{hr}$ the cooldown curve may be tabulated thus

Θ' (hours)	0	2.0	4.0	6.0	8.0	10.536
Q/Q_o	1.0	0.980	0.961	0.942	0.923	0.900

Using equation (9), the duration of recovery, Θ_2, is

$$\Theta_2 = \frac{1}{\eta_s} \ln \left(\frac{\frac{Q_1}{Q_o} - \frac{1}{\eta_s}\frac{q_e}{Q_o}}{1 - \frac{1}{\eta_s}\frac{q_e}{Q_o}} \right) \tag{12}$$

The two ways to shorten the duration of recovery are:

1) by reducing η_s and/or

2) by increasing q_e.

The duration between the thermostat cutout and the subsequent thermostat cutin, the duration of cooldown θ_1, may be derived from the cooldown equation,

$$\theta_1 = \frac{-1}{\eta_s} \ln \left(\frac{Q_o}{Q_1}\right). \qquad (13)$$

One way to lengthen θ_1 is to reduce η_s, by adding insulation.

There are many reasons why the above analysis does not describe the water heater systems adequately, among them:

1. This analysis does not lend itself to the study of the dynamic behavior of draw.

2. The overall conductance is not a constant, but rather a complex function of temperatures.

3. The heat capacity of the composite walls of the water heater is not taken into consideration.

For these reasons and also in order to study the effects of the distribution pipe, a more elaborate model and computer code is developed.

This computer simulation model, called NBS/ESDS (Energy Storage and Distribution System) developed at the National Bureau of Standards (NBS), can be used for studying the operation and designs of storage-type hot water systems. A separate users' manual (3) has been prepared and includes a description of the subroutines, a listing of the computer program as implemented on the NBS UNIVAC 1108 and the requirements for the actual preparations of input data set.

III. Modeling of the Water Heater and Its Distribution Pipes

The water heater currently considered is assumed to be an electric water heater having cylindrical, axisymmetrical storage tank. Each layer of its composite wall is assumed to form a right cylinder. Cold supply water is assumed to enter the tank from the bottom and hot water withdrawn from the top. The distribution pipe consists of one single run of pipe with no branches and has constant diameter and thickness.

The basic elements of the modeling for the water heater and the associated distribution pipes consists of a conduction model, a pipe flow model, and a series of auxiliary models.

The conduction model is used to calculate the unsteady conduction heat transfer for an axisymmetric solid body. It is used to simulate the heat transfer of a water heater in which the water is temporarily considered a solid.

The same conduction model is also used to calculate the heat transfer of a distribution pipe during the non-draw periods. The water inside the pipe is also temporarily considered as a solid.

The pipe flow model is used to calculate the heat transfer from the distribution pipe during the periods of the draw. The governing equations are two simultaneous (for single wall pipe) partial differential equations involving only first order terms in terms of time and longitudinal distances. The water temperature is considered to be a function of time and the longitudinal distances and not of the radial distances. The water is considered to be flowing at a velocity uniform across the entire pipe inner cross-section. Therefore, the distribution pipe is modeled alternately with a conduction model and a pipe flow model depending on the mode of pipe operation. During the non-draw periods the former is used; during the draw periods the latter is used. Since the non-draw periods and the draw periods are interspersed in a 24 hour cycle, the solution of the conduction model forms the initial conditions of the subsequent pipe flow model; and the solution of the pipe flow model forms the initial conditions of the subsequent conduction model.

The other important features incorporated in the model are:

1. Buoyancy stacking. It is used to simulate the process of thermal stratification, in which water elements at higher temperatures float to a higher elevation and water elements at lower temperatures sink to a lower elevation in the storage tank of the water heater. In vertical sections of the distribution pipes, the same type of stratification occurs.

2. Heating elements. The heat source terms are included in the heat equation of the conduction model to simulate the heating elements.

3. Draw schedule. Depending on the flowrate of a draw cycle and the size of the time step of the water heater conduction model, different algorithms are used to simulate a draw of hot water.

A. Conduction Model

There are two basic approaches for solving the transient heat conduction equation governing the heat transfer in a water heater which is cylindrical and axisymmetrical.

1. The first approach is to apply a finite difference technique to the heat transfer equation (14). Essentially, the scheme involves approximating the various partial derivatives with their discrete counterpart:

$$\frac{1}{\alpha}\frac{\partial T}{\partial \theta} = \frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial Z^2} \qquad (14)$$

where $\frac{\partial T}{\partial r}$ may be approximated by $\frac{T_{i+1,j} - T_{i,j}}{\Delta r}$,

$\frac{\partial^2 T}{\partial r^2}$ may be approximated by $\frac{T_{i+1,j} - 2T_{i,j} + T_{i-1,j}}{\Delta r^2}$,

$\frac{\partial T}{\partial \theta}$ may be approximated by $\frac{T_{i,j}^{n+1} - T_{i,j}^{n}}{\Delta \theta}$ and

α is the thermal diffusivity of the medium.

This approach has the advantage of being quite straightforward for a homogeneous body of simple geometry. The disadvantage is its awkwardness to handle a body of mixed materials with complex geometry.

2. The second approach is to divide the physical model into a number of small elements (nodes). A conservation of energy equation is written for each of the nodes, relating the temperature change in time in a central node with temperatures in its neighboring nodes. Denoting T_0 as the temperature of the central node and T_1 through T_4 as those of its neighboring nodes, Figure 2, then the conservation of energy equation for node 0 may be written as

$$C_1(T_1-T_0)+C_2(T_2-T_0)+C_3(T_3-T_0)+C_4(T_4-T_0)=(T_0'-T_0)C_5/\Delta\theta \qquad (15)$$

where the coefficients, C's are functions of thermal properties and geometrical parameters and $(T_0'-T_0)$ is the temperature change of the node 0 in a time step of $\Delta\theta$. This approach is versatile for handling a model of mixed materials and complex geometry since any change from node to node, of properties, volumes, areas or heat transfer modes can easily be incorporated into the conservation of energy equation. Therefore, this approach is more suitable for the modeling of water heaters and was chosen for this study.

For time-advancing, either an explicit or an implicit method may be used. A detailed discussion of the problems of convergence and stability and other merits of each method is beyond the scope of this report and has been presented elsewhere (4). The implicit method is used because it provides unconditional stability regardless of the size of the nodes or the time step. The two different methods are summarized briefly in the following paragraphs.

If an explicit method is used then all the unprimed temperatures in equation (15) are known because the starting point of the time-advancing is the initial condition, which is given. The primed temperature, evaluated at one time step in the future, is the only unknown. Therefore, the temperature of a node one time step in the future may be calculated using equation (15) if the present temperature in the five nodes are known. Writing a similar equation for each of the nodes, the temperature one time step in the future for all the nodes may be found if the present temperature and the boundary conditions are known. On the other hand, if an implicit method is used, then all the temperatures on the left-hand side and T_0' on the right-hand side of equation (15), are evaluated at a future time, while T_0 on the right-hand side is evaluated at the present time. Unlike in the explicit method T_0' cannot be evaluated directly using this one equation, since there are 5 unknowns in this one equation. Writing one conservaton of energy equation for each of the nodes results in a system of algebraic equations whose coefficient matrix consists of five diagonal non-zero terms. With given boundary conditions, there are an equal number of unknowns and equations. The temperature of all the nodes may be calculated by solving this system of equations.

Solving this system, however, may be laborious and time-consuming. An alterating direction method (5) may be used resulting in solutions for two sequential tri-diagonal systems of equations. Equation (15) is then written as two equations:

$$C_1(T_1^*-T_0^*)+C_2(T_2^*-T_0^*)+C_3(T_3-T_0)+C_4(T_4-T_0)= C_5(T_0^*-T_0)/(1/2)\Delta\theta \qquad (16)$$

and

$$C_1(T_1^*-T_0^*)+C_2(T_2^*-T_0^*)+C_3(T_3'-T_0')+C_4(T_4'-T_0')= C_5(T_0'-T_0^*)/(1/2)\Delta\theta \qquad (17)$$

In the above equations the unsuperscripted terms are temperatures evaluated at the present, the starred terms at one-half time-step in the future, and the primed terms at one full time step in the future. Equation (16), which for this case is implicit in the r-direction only and written out fully for all the nodes, may be used to find the temperature distribution at one half time step in the future. It involves solving a system of equations whose coefficient matrix consists of only three diagonal non-zero elements (hence tridiagonal matrix). Equation (17), which is implicit in the z-direction only, may be solved using the same method, since the starred temperatures have already been calculated from equation (16).

B. Verification of the Conduction Model

In order to verify the basic numerical techniques of the NBS/ESDS program, a test problem was solved analytically as well as using ESDS. The test problem is to solve the transient temperature distribution of a cylindrical rod 2.0 units in diameter and 2.0 units in height, Figure 3. At the initial time its temperature is T_i. From that time on, the rod is so submerged in a fluid that the film coefficients on the end surfaces, h_z, are 1.0; and the film coefficients for the lateral surfaces, h_r, are 0.5. Mathematically the problem is

$$\frac{\partial T}{\partial \Theta} = \frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial Z^2} \qquad (18)$$

with initial and boundry conditions of

I. C. $T = T_i$ $0<r\leq 1$, $0\leq z\leq 2$, $\Theta = 0$

B. C. $h_z = 2.0$ $0\leq r<1$, $z = 2$, $\theta\geq 0$

$h_z = 2.0$ $0\leq r<1$, $z = 0$, $\theta\geq 0$

$h_r = 0.5$ $r = 1$, $0\leq z\leq 1$, $\theta\geq 0$

The analytical solution of equation (18) is given in the literature, e.g. (6) and is plotted in Figure 3. The temperature distribution for the same problem was calculated by running NBS/ESDS and is also plotted in Figure 3. The agreement indicates the correctness of the numerical scheme.

C. Pipe Flow Model

As the hot water flows in the distribution pipe, it losses heat through the walls of the pipe to the environment. The temperature of the water is a function of time, Θ; and distance, z, i.e., $T=T(z,\Theta)$. Problems of this nature have been studied by many authors (for example, 7, 8, 9, 10). In general, under the following assumptions:

1. The velocity and temperature of the fluid do not vary in the radial direction (piston or plug flow).

2. The temperature of the fluid is a function of time and of the distance from the pipe entrance.

3. Both the diameter and the wall thickness of the pipe are constant throughout the system.

4. Each pipe layer is made of material of constant properties.

5. Heat transfer in the radial direction is much greater than that in the longitudinal direction.

The governing equations for the temperatures of the hot water and the two layers of the pipe shown in Figure 4 are as follows:

$$\pi r_i^{\ 2} c_f \rho_f \frac{\partial T}{\partial \theta} = \dot{m} c_f \frac{\partial T}{\partial x} - 2\pi r_i U_i(-u+T) \qquad (19)$$

$$\pi(r_o^{\ 2} - r_i^{\ 2})\ \rho_u c_u \frac{\partial u}{\partial \theta} = 2\pi r_i U_i(T-U) + 2\pi r_o U_o(y-u) \qquad (20)$$

$$\pi(r_s^{\ 2} - r_o^{\ 2})\ \rho_y c_y \frac{\partial y}{\partial \theta} = 2\pi r_o U_o(u-y) + 2\pi r_s U_s(T_{amb}-y) \qquad (21)$$

In the above equation set, T, u, y are the temperatures of water, the inner and outer layer of the pipe, respectively. The r is the various radii of the pipe; c the specific heat; ρ the density; $\dot{m}$ the mass flow rate; U the unit overall conductance; and T_{amb} the ambient temperature. Note that in equation (19) the term on the left hand side represents the rate of change of internal thermal energy locally. The first term on the right hand side represents the rate of change of internal thermal energy due to flow. The second term on the right hand side represents the heat transfer to the wall from the water. The internal thermal energy reduction (or increase) of an element of the hot water is due to the flow and heat transfer with the wall. Equations (20) and (21) indicate that the rate of change of internal thermal energy locally is due to heat transfer in the radial direction with the neighboring medium.

To calculate the temperature distribution of pipe flow, an explicit method is used. The explicit method involves a staggered nodal system, Figure 5.

For equation (19) at nodal point 2,

$$\frac{\partial T}{\partial \theta} \cong \frac{T_2' - T_2}{\Delta\theta} \qquad (22)$$

where T_2 is the temperature at 2 at the current time, θ; T_2' the temperature at 2 at a future time, $\theta+\Delta\theta$. And

$$\frac{\partial T}{\partial x} \cong \frac{T_2 - T_1}{\Delta x} \qquad (23)$$

where Δx is the distance betewen nodal points 1 and 2. And

$$(T - u) \cong \left(\frac{T_1 + T_2}{2} - u_A\right) \qquad (24)$$

The staggered nodal system aligns the (T–u) and ∂T/ ∂x.

For the inner layer of the pipe described by equation 20,

$$\frac{\partial u}{\partial \Theta} \cong \frac{u_A' - u_A}{\Delta\Theta} \tag{25}$$

$$(T-u) \cong \left(\frac{T_1 + T_2}{2} - u_A\right)$$

and

$$y - U = (y_D - u_A) \tag{26}$$

And similarly for the outer layer of the pipe described by equation 21,

$$\frac{\partial y}{\partial \Theta} \cong \frac{y_D' - y_D}{\Delta\Theta} \tag{27}$$

$$u - y = (u_A - y_D) \tag{28}$$

and

$$T_{amb} - y = (T_{amb} - y_D) \tag{29}$$

As a result of the above approximation or substitution, equation (19) may be rewritten as

$$T_2' = T_2\,(1-a-\tfrac{1}{2}b) + T_1(a-\tfrac{1}{2}b) + bu_A \tag{30}$$

where $a = \dfrac{\dot{m}\,\Delta\,\Theta}{\pi r^2 \Delta x \rho_f}$, $b = \dfrac{2U_i\Delta\Theta}{r_i\rho_f c_f}$

Equation (20) may be rewritten as

$$u_A' = U_A(1-c-d)f\tfrac{1}{2}\,c\;T_2 + \tfrac{1}{2}\,c\;T_1 + dy_D \tag{31}$$

where $c = \dfrac{U_i\;A_i\;\Delta\Theta}{\pi(r_o^{\,2}-r_i^{\,2})\Delta x\rho_u c_u}$, $d = \dfrac{U_o A_o\;\Delta\Theta}{\pi(r_o^{\,2}-r_i^{\,2})\Delta x\rho_u c_u}$

with $A_i = 2\,\pi\,r_i\,\Delta\,x$,

and $A_o = 2\,\pi\,r_o\,\Delta\,x$.

Equation (21) may be rewritten as

$$y_D' = y_D(1-p-g) + pu_A + gT_{amb} \tag{32}$$

where $$p = \frac{U_oA_o \; \Delta\Theta}{\pi(r_s^2-r_o^2)\Delta x \rho_y c_y}, \quad g = \frac{U_sA_s \; \Delta\Theta}{\pi(r_s^2-r_o^2)\Delta x \rho_y c_y}$$

The overall unit conductance, U_i, is a function of the water side film coefficient, h_i. Likewise, U_s is a function of the air side film coefficient, h_s.

Using equation (30), and for given T_2 and boundary condition T_1, T_2' is calculated. Similarly, T_3' and the rest of the temperature distribution of the fluid nodes are calculated.

Using equation (31), and for given u_A, y, T_2 and T_1, u_A' is calculated.

Using equation (32), and for given y_D. u_A, and T_{amb}, y_D' is calculated.

Dusinberre (8) discussed the criterion for selecting the time-step, $\Delta\theta$. The criterion is that all the coefficients of temperatures on the right hand side of equations (30), (31), and (32) be positive. In case of equations (31) and (32) the criterion specifies that

$$\frac{(U_iA_i+U_oA_o) \; \Delta\Theta}{\pi(r_o^2-r_i^2)\Delta x \rho_u c_u} < 1$$

or $$\Delta\Theta < \frac{\pi(r_o^2-r_i^2)\Delta x \rho_u c_u}{(U_iA_i + U_oA_o)} \tag{33}$$

In case of thin copper pipes as commonly used, $\Delta\theta$ has to be extremely small to satisfy the criterion. To circumvent the expensive computation involving extremely small time-step, the equations (20) and (21) are reconsidered. For pipe flow in thin-walled pipes with high conductances the left hand side terms are small in comparison with the right hand side terms so if the left hand side terms are neglected, equations (20) and (21) may then be rewritten as,

$$r_iU_i(T-u) + r_oU_o(y-u) = 0 \tag{34}$$

$$r_oU_o(u-y) + r_sU_s(T_{amb}-y) = 0. \tag{35}$$

The temperatures of the wall layers may then be calculated for the nodes shown in Figure 4, using the following substitution

$$(t-u) = \frac{T_1+T_2}{2} - u_A, \tag{24}$$

$$(y-u) = (y_D-u_A) \tag{36}$$

and

$$T_{amb}-y = T_{amb}-y_D, \tag{37}$$

The temperatures may be calculated,

$$T_2' = T_2(1-a-1/2\ b) + T_1(a-1/2\ b) + bu_A \tag{30}$$

$$u_A' = \frac{r_iU_i\left(\frac{T_1'+T_2'}{2}\right) + r_oU_oy_D}{r_oU_o+r_iU_i} \tag{38}$$

$$y_D' = \frac{r_oU_oU_A' + r_sU_sT_{amb}}{r_oU_o + r_SU_s} \tag{39}$$

The procedure is

1) calculate T' first using the transient heat transfer equation 30;

2) calculate u_A, using the fluid temperatures as calculated above, the current y_D and using equation 38;

3) calculate y'_D using the temperature of inner wall as calculated above in equation 39.

D. Film-Coefficients

The film coefficients, h, for both the water heater model and the pipe model (conduction and flow) consist of two components: convective (h_c) and radiant (h_r), or

$$h = h_c + h_r.$$

The convective film coefficient as modeled in the NBS/ESDS may be one of the following:

1) given constant,

2) a calculated value which is a function of temperatures as,

a) in case of turbulent natural convection,

$$h_c = C(T_s-T_a)^{1/3} \tag{41}$$

where,

T_s = the surface temperature

T_a = the ambient temperature

b) in case of laminar natural convection,

$$h_c = C(T_s - T_a)^{1/4}. \qquad (42)$$

The radiant film coefficient, expressing the temperatures is in terms of absolute temperatures, calculated as:

$$h_r = F_{1-2}\, \sigma \frac{(T_s^4 - T_a^4)}{(T_s - T_a)} \qquad (43)$$

where,

F_{1-2} = the gray body shape and emissivity factor,
σ = the Stefan-Boltzmann constant.

Two assumptions are made in determining F_{1-2}.

1) In case of the pipe and the lateral surfaces of the water heater, F_{1-2} is calculted as that of two concentric cylinders of infinite length,

$$F_{1-2} = \frac{1}{\frac{1}{\varepsilon_1} + \frac{r_o}{r_a}\,\frac{1-\varepsilon_2}{\varepsilon_2}} \qquad (44)$$

where,

ε_1 = the emissivity of the concerned cylinder surface

ε_2 = the emissivity of the walls of the room where the cylinderis located.

r_a = the characteristic radius of the room

r_o = the radius of the cylinder.

2) In case of the top and the bottom surfaces of the water heater, F_{1-2} is calculated as that of two infinite parallel plates,

$$F_{1-2} = \frac{1}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} \qquad (45)$$

The procedure of iteration is:

a) calculate the film coefficients with the current surface temperatures and the ambient temperature with equations 41, 42 and 43;

b) calculate the temperature distribution according to the methods described in the conduction model or the pipe flow model with equations 30, 38 and 39;

c) calculate the surface temperatures from the temperature distribution of the nodes at the appropriate boundry.

E. Draw Model

The draw model is dependent on the node size of the water region and the time step. Different algorithms are used depending on the draw volume of one single time step.

1) Option 1. Draw in one time step greater than that contained in each horizontal section of the tank (one row of horizontal nodes). A r-z cross-section of a tank full of water before draw is shown in Figure 6(a). The axis of the cylindrical tank coincides with the left-hand edge of the cross-section. Note that the spacings of the grid line are not uniform in order to make all the nodes equal volumed. In Figure 6(b) the bottom portion of the stored water is displaced by cold supply water (cross-hatched), while the volume of draw is shown on top of the cross-section.

Figure 6(c) shows the temperature distribution of one typical column of nodes after one $\Delta\Theta$ of draw. The bottom most node (node 1) is cross-hatched. The water there is displaced by the supply water and the new temperature is the temperaure of the supply water. This process is indicated by $T_1 \leftarrow T_s$. The node 2 has a new temperature which is a combination of the old temperature of the node 1, T_1, and the supply temperature T_s. This process is indicated by $T_2 \leftarrow (T_1/T_s)$. This algorithm is called a mixed shift and the actual calculation is

$$T_2 \leftarrow \frac{V_s \times T_s + (V - V_s) \times T_1}{V} \qquad (46)$$

where V is the volume of each of the nodes. V_s is the volume of supply water that has displaced the hot water in the node 2. The node 3 has a new temperature $T_3 \leftarrow (T_2/T_1)$, etc. The delivered water from this column has a temperature which is a combination of T_6 (whole node) and T_5 (partial node).

Figure 6(d) shows the temperature distribution at the end of the second $\Delta\Theta$ of a continuous draw. As in the first $\Delta\Theta$ of the draw the bottom portion of the column is displaced by the supply water (cross-hatched), while the cross-hatched area of Figure 6(c) is now pushed upwards and is now diagonal hatched. This is because the old supply water which has been resident in the water tank for one $\Delta\Theta$ and has been subject to heat transfer in the tank. The procedure to calculate the new temperature distribution is same as in the first $\Delta\Theta$.

2) Option 2. Draw in one time step greater than the volume of one node but less than a section of the water region. Figure 7(a) shows the water region prior to a draw. Figure 7(b) shows the temperature distribution at the end of the first $\Delta\Theta$ of a draw. The cross-hatched area indicates the nodes or partial node where the temperatures have been replaced by that of the supply water. The delivered water, which is considered as outside the water region is shown at top of Figure 7(b). In the first two columns of nodes (counting from the axis of the water region), the temperature of both of the first nodes are replaced by the supply water temperature, $T_1 \leftarrow T_s$. The temperatures of both of the second nodes are replaced by the old temperature of the first nodes, $T_2 \leftarrow T_1$, This algorithm is called a simple shift as mentioned previously. In the third column, the temperature of the first node is also replaced by the temperature of the supply water. The second node of the same column, however, is merely partially displaced by the supply water. It's temperature, then, is replaced by the combination of the old temperature of the first node and the supply water, $T_2 \leftarrow (T_1/T_s)$. The new temperatures of all the nodes in the third column except the first node are calculated using the mixed shift algorithm.

Figure 7(c) shows the temperature disribution at the end of the second $\Delta\Theta$ of a continuous draw. This time the first nodes of the fourth and the fifth columns are filled with the supply water (simple shift). The first node of the first column is also filled with the supply water, the second node of the first column is partially filled with the supply water (mixed shift). In the meantime, the supply water residing in the water region for one time step has been subject to local heat transfer.

Figure 7(d) shows the temperature distribution at the end of the third $\Delta\Theta$ of the continuous draw. The same procedure is used to calculate the current temperature distribution. The supply water residing in the water region for two $\Delta\Theta$ is shown dotted. The supply water residing in the water region for one $\Delta\Theta$ is shown diagonal hatched indicating continuous heat transfer during the three $\Delta\Theta$ draws.

3) Option 3. Draw of one time step less than the volume of one node. Figure 8(a) shows the temperature distribution of the water region at the end of a time step during a draw. This option has the same initial temperature distribution prior to draw, Figure 7(a). The first node of the first column now has a new temperature which is a combination of the old T_1 and T_s, $T_1 \leftarrow (T_1/T_s)$ (mixed shift). At the end of the second $\Delta\Theta$ of a continuous draw, Figure 8b, the first node of the second column now has a new temperature as a result of a mixed shift. In the meantime, the water in the first node of the first column has undergone local heat transfer.

If the draw is continuing for more than five $\Delta\Theta$, under the current grid system, the first nodes of all the columns will have been subject to a mixed shift. At the end of the sixth $\Delta\Theta$, the second node of the first column will have undergone a mixed shift for the calculation of its new temperature. The number of $\Delta\Theta$ in a draw schedule using Option 3 may not exceed the total number of water nodes (20) in the case of the grid system shown in Figure 8.

Two approaches may be used to accommodate longer durations of a draw with small flow rate: i) using a finer mesh or grid system in the water region; and ii) reducing the time step.

Erroneous calculations will result under the current modeling if the first node of the first column is used again at the sixth $\Delta\Theta$. The cause for error may be illustrated as follows:

a. Assume the temperature of the first node of the first column prior to draw as T_1, Figure 9; its volume as 1.0 the draw in one $\Delta\Theta$ as a fraction of the nodal volume as (x).

b. At the end of one $\Delta\Theta$ of a draw the volume of hot water is (1-x), Figure 9(a); the volume of supply water is (x); the new temperature of the node then:

$$T_1^{(1)} = (1-x)T_1 + xT_s.$$

c. At the end of a second $\Delta\Theta$ of a continuous draw, Figure 9c, the volume of hot water is (1-2x); the volume of supply water residing in the node for one time step, (x); the volume of fresh supply water, (x); the new temperature of the node then:

$$T_1^{(2)} = (1-2x)T_1 + xT_s' + xT_s.$$

This calculation would be correct if the old water remained at T_1 during the two time steps, but it changed as explained below.

There are two reasons why nodal temperatures during a draw should not be calculated as above. i) it requires additional computer memory (for T_1, etc.); and, ii) it requires heat transfer calculation for a fractional node. Heat transfer calculation for a fractonal node is not done by definition under the current context.

Under the current modeling, when the algorithm mixed shift is performed,

$$T_1 = (1-x)T_1 + xT_s$$

The old water temperature is lost, because the quantity on the right-hand-side of the equation has been stored in the location "T_1". Therefore, when the mixed shift is performed on the node again

$$T_1 = (1-x)T_1 + xT_s$$
$$= (1-x)[(1-x)T_1+xT_s] + xT_s,$$

The result is different from the correct calculation as in (c) above.

F. Buoyancy Stacking

A sorting algorithm is used to generate a thermal stratification both in the water region of the tank or in vertical sections of distribution pipes during standby. The sorting algorithm merely reorders a series of numbers to a decreasing order. For example, in processing the temperatures of the nodes in a column , the sorting algorithm will reorder the numbers in a decreasing order, so that the highest temperature will be associated with node at the top of the column, the lowest at the bottom.

To model the heat loss due to a stand pipe for the purpose of simulating a heat trap or the lack of it, the conservation of energy has to be considered in addition to the buoyancy stacking. The first vertical segment of pipe, sometimes referred to as a stand pipe, connecting the distribution pipe network with the water heater causes heat leaks due to boyancy effects. These heat leaks may be reduced by the installation of heat traps.

Shown in Figure 10(a) is a temperature distribution of a column of the water in the tank and a contiguous stand pipe. The sorting algorithm reorders the temperatures in such a way so that $TP_2' > TP_1' > T_4' \ldots > T_1'$. In order to satisfy the conservation of energy the following adjustment is made. Let e_p be the energy of the stand pipe before the buoyancy stacking; e_w the energy of the column of water before the buoyancy stacking; e_p' and e_w' the corresponding energy after the buoyancy stacking. The total energy of the system is e_t, or

$$e_t = e_p + e_w \qquad (47)$$

where $$e_p = \sum_{i=1}^{k_p} \rho c \; VP_i \; TP_i \qquad (48)$$

and $$e_w = \sum_{i=1}^{k_w} \rho c \; V_i \; T_i \qquad (49)$$

In the above equations ρ and c are the density and specific heat of water, respectively; VP_i, TP_i the nodal volume and temperature of the stand pipe, respectively; V_i and T_i the nodal volume and temperature of the water column, respectively; k_p and k_w are the number of nodes in the stand pipe and the water column, respectively.

The energy of the stand pipe and the water column after the buoyancy stacking, respectively, are:

$$e_p' = \sum_{i=1}^{k_p} \rho c \; VP_i \; TP_i' \qquad (50)$$

$$e_w' = \sum_{i=1}^{k} \rho c \; V_i \; T_i' \qquad (51)$$

The energy of the water column after the thermal stacking should be

$$e_w'' = e_t - e_p' \tag{52}$$

The difference, because the water nodes and the pipe nodes have different volumes, is:

$$e_w'' - e_w' \tag{53}$$

In order to reduce the above difference to zero—satisfying the conservation of energy principle—an adjustment to the water column temperature is made. The adjustment, as calculated below, is made only to a number of nodes at the bottom of the water column. This number equals to the number of nodes in the stand pipe,

$$\Delta T = \frac{e_w'' - e_w'}{\sum_{i=1}^{k_p} \rho c \; VP_i} \tag{54}$$

Then

$$T_1'' = T_1' + \Delta T$$

$$T_2'' = T_2' + \Delta T$$

$$T_{kp}'' = T_{kp}' + \Delta T$$

The T_1'', T_2'', ..., T_{kp}'', T_{kp+1}', T_{kp+2}', ..., T_k' are then the correct, buoyancy-stacked, temperature distribution of the water column.

IV. Parametric Studies

A. Description of the Water Heater and the Distribution Pipe

A commercially available "52 gallon" electric water heater was modeled for these studies. The actual capacity is 191 L (49.8 gal.). The exterior shell measures 1.45 m (57.2 in) in height and 0.58 m (22.9 in) in diameter. The shell is made of 21 gage sheet carbon steel and consists of a right cylinder with a top and a bottom cover. The cylindrical water tank with convex top and bottom is made of 12 gage sheet carbon steel having an interior anti-corrosion cement lining 1.8 cm thick. A fitting for connecting a relief value is located at the center of the top. Three legs support the water tank on the bottom of the exterior shell. There are two heating elements, two thermostats, water outlet and inlet, junction box and wiring located on the side walls of the water tank. In this particular water heater there is also a built-in heat trap connected to the hot water outlet. The annular gap varying from 2.5 cm (1.0 in) to 6.0 cm (2.4 in) is filled with fiberglass insulation.

A typical residential hot water distribution piping system consists of one main line and a few branch lines. In the current study, the distribution piping system is assumed to consist of one single pipe, 6.71 m (22 ft.) long, 1.68 cm (0.6619 in.) OD, 1.60 cm (0.631 in.) ID. This pipe has two vertical sections and one horizontal section.

B. Modeling of the Water Heater

1. Basic Water Heater (Water Heater A)

The exterior shell and the water tank of the water heater are considered as right cylinders. The actual outer diameters of the water tanks are used, the heights are derived from the measured volumes of the tanks. The steel walls and the insulation are taken to have uniform thicknesses. Their thermal properties are considered constant. Different thermal conductivities for insulation are used, depending on the temperature range any particular layer will experience to account for the thermal conductivity variation as a function of temperature. The thermal conductivity of an insulation layer is evaluated at the average of its temperature range during the cooldown period of the water heater. Water heater fittings and leakages are not independently considered but are lumped with the mean thermal conductivity of the insulation which is summarized in Table 1. Internal water current has been considered by using a water thermal conductivity 100 times higher* than the actual value to account for internal mixing. Radiative and convective heat transfer parameters are given in Table 1. An initial condition of 71.1^{o}C (160^{o}F) is used for all the nodes while a constant ambient temperature of 21.1^{o}C (70^{o}F) together with the surface heat transfer coefficients are used to describe the boundary conditions.

In modeling the electric water heater, 207 nodes are used, Figure 11. It was established in earlier work (2) that 104 nodes were sufficient for accurate modeling. The 207 nodes were chosen to provide sufficiently small nodal volumes of the water region to accommodate small flowrate during the draw. In the water region the nodes have equal volume. There are 10 grid lines in the radial direction and 24 grid lines in the axial direction [(10-1)x(24-1)=207]. There are 6 radial grid lines in the water region, two in the cement lining, two in the tank walls, two in the insulation and two in the exterior shell; among these 14, four are common grid lines [14-4=10] which constitute boundries between layers. From the bottom to the top of the water heater there are two axial grid lines in the exterior shell bottom, two in the insulation, two in the tank bottom, two in the lower cemenet lining,, 16 in the water, two in the upper cement lining, two in tank top, two in the upper insulation, and two in the upper exterior shell top; among the 32, 8 are common [32-8= 24].

A time step of .01 hours is sufficient for accurate modeling of the water heater (2).

*Footnote: This value which can easily be adjusted was chosen arbitrarily in the absence of experimental data on the internal mixing.

For the exterior shell of this water heater, the product of Grashof and Prandtl number, X, exceeds 10^9; thus, the film coefficient for the vertical surfaces with turbulent flow is expressed as in equation 41:

$$h_o = C_o(T_o - T_a)^{1/3},$$

the film coefficient for the horizontal top surface with laminer flow is detemined from equation 42 as:

$$h_t = C_t\left(\frac{T_t - T_a}{D}\right)^{1/4}, \text{ and}$$

the film coefficient for the horizontal bottom surface is also for laminer flow and determined from equation 42 as:

$$h_b = C_b\left(\frac{T_b - T_a}{D}\right)^{1/4}.$$

The coefficients of the above correlations as used in the NBS/ESDS are summarized in Table 1.

Two 4.5 kW heating elements—same number and ratings as in the actual water heater—are modeled. Each of them is considered as an evenly distributed heat source in a row of four contiguous nodes, the second through the fifth nodes from the center of the water heater, Figure 11. the upper heating element is located at the 15th row of nodes from the bottom of the water heater which is 86.1 cm (34.0 in) from the bottom of the water tank; the lower heating element at the eighth row of nodes is 34.3 cm (13.5 in) from the bottom of the water tank.

Two thermostats are modeled. Both thermostats are located on the outer surface of the water tank. The upper thermostat is located at the eighteenth node, 93.5 cm (36.8 in), from the bottom of the water heater and the lower thermostat at the eighth node, 34.3 cm (13.5 in), from the bottom of the water heater. The upper thermostat controls the upper heating element and the lower thermostat the lower heating element. The thermostats and the heating elements are arranged in such a way that only one heating element operates at one time. The upper limit for both of the thermostats is 71.1°C (160°F) and he lower limit 68.3°C (155°F). If the water temperature is so low after a heavy draw or during a startup that both thermostats sense temperature below the lower limit of the thermostats, the upper thermostat takes precedence. The thermostats sense only the temperature of the outer tank wall. Since there is a cement lining with low thermal conductivity between the water and the tank wall, the thermostat does not respond directly to the water temperature. For example, the lower thermostat is located at the tank wall node of the eighth row of nodes.

Counting from the center of the water heater, the second through the sixth are the water nodes, the seventh the cement node and the eighth the tank wall node. In a recovery mode at a certain time the temperature of the sixth node (water) reaches 71.1°C (160°F), the thermostat node, however, does not reach this temperature until a few times steps later. In fact the water node may reach 73.9°C (165°F) before the thermostat acts to cutout the heater. The reverse of this occurs before the cutin point of the thermostat.

2. Insulated Water Heater (Water Heater B)

The same 207 node model is used to study an electric water with an extra 2.5 cm (1 in) of insulation applied to the exterior shell (top and side walls). The only difference from water heater A is that the steel casing regions on the top and on the side walls of the model are replaced with insulated regions. The presence of the thin layer of the casing has very little effect on the heat transfer properties of the water heater model. The same film coefficients and mean thermal conductivity for water heater A insulation are used, while the extra insulation has a thermal conductivity compatible with its type.

C. Modeling of the Distribution Pipe

1. Distribution Pipe (Pipe A)

The distribution pipe, as shown in Figure 12 is 671 cm (264 in) long. It consists of two vertical sections of 91.4 cm, (36 in) and one 488 cm (192 in) horizontal section. The radii given in section IVA are used. The copper walls are taken to have uniform thickness. Their thermal properties are considered constant. Internal water mixing during standby in the horizontal section of the pipe is considered to be minimal. The standard text book value of the thermal conductivity is used for the water. In the vertical section of the pipe, the process of the buoyancy stacking is assumed to place.

Film Coefficients. Film coefficients for the pipe are calculated from correlations recommended by McAdams (11) for different value of X--the product of Grashof and Prandtl numbers.

a) Vertical Section

Preliminary studies indicate that the maximum surface temperature for bare pipe is about 70C (158°F) and the corresponding ψ ($=\rho^2 g \beta c_p/\mu k$) is approximately 10^6. The corresponding $X = 7.04 \times 10^8$ for a 61 cm (24 in) vertical pipe section. Therefore, for laminar natural convection (from equation 42)

$$h_v = 1.42\left(\frac{\Delta T}{L}\right)^{0.25} \qquad (54)$$

where h_v is in watt/m^{2}K, L in m, ΔT the temperature difference between pipe surface and ambient in K. For insulated vertical pipes, the maximum surface temperature is approximately 46.1^{o}C (115^{o}F), $\psi = 1.2 \times 10^{6}$, and $X = 4.3 \times 10^{8}$ for 61 cm (2 ft.) sections. Therefore, equation (54) is also applicable for insulated vertical pipe sections.

b) Horizontal Sections

Preliminary studies indicate that the maximum surface temperature for bare pipe is 66.7C (152^{o}F). The corresponding ψ is 10^{6}, $X = 5 \times 10^{4}$, therefore, for laminar natural convection from equation (42).

$$h_h = 1.32\left(\frac{\Delta T}{D}\right)^{0.25} \qquad (55)$$

where D is the diameter of the pipe. For the insulated horizontal pipe, equation (55) may also be used.

c) Combination Vertical/Horizontal Sections

Where a node consists of a horizontal and vertical segment, a prorated film coefficient may be used.

d) Water Side Film Coefficient

McAdams (11) recommends for calculation of the water side film coefficient for pipe flow the following formula:

$$h_w = 1057 \left[1.32 + .018\ T_b\right] \frac{V^{0.8}}{D^{0.2}}$$

where h_w is in terms of watt/m^{2}K; T_b, the bulk fluid temperature in C; V, the fluid velocity, in m/sec; D, the inside diameter of pipe, in m. Preliminary studies gave the following typical values.

T_b = 48.9^{o}C (120^{o}F)

V = 0.17 m/sec (0.55 ft/sec), and

D = 0.016 m (0.63 in)

The typical value of h_w, then is 1288 Watt/M^{2}K (227 Btu/hr ft^{2} ^{o}F) which was used in the modeling.

As discussed previously, during a non-draw cycle the distribution pipe is considered as a conduction model. The water inside the pipe is considered as solid. Only a model of buoyancy stacking is used to simulate the fact that in vertical sections, the warmer portion of the water floats to the top and the colder portion sinks to the bottom. During a draw cycle, the pipe flow model is used to model simulate the distribution pipe, i.e., a fluid flows inside the pipe. In the latter case a fluid side film coefficient is used.

2. Insulated Distribution Pipe (Pipe B)

The nodal diagram of the insulated pipe model is same as that of the basic pipe model except that the radii are different. The radius of the water region, r_i, (Figure 5) remains the same. The radius of the copper pipe, r_o, now is 1.68 cm (0.661 in). The radius of the insulation (polychloroprene) r_s, is now 2.79 cm (1.1 in). The properties used for polychloroprene, given in Table 1, are taken from (12).

D. Modeling of the System

1. Water Heater-Pipe Connection

The water heater pipe connection is shown in Figure 12(b). The water connection is made at the second node of the top row of the water region. The pipe connection is made at the second node of the top of the tank wall region. Mathematically, the contiguous water temperature becomes the boundary condition for calculating pipe water temperature in both of the pipe flow model and the pipe conduction model. Similarly, the temperature of the contiguous tank wall region becomes the boundary condition in calculating pipe wall temperature in the pipe conduction model. The heat transfer in the pipe conduction model at the connection, however, is negligible.

2. Heat Traps

The lack of a heat trap is modeled by the modified buoyancy stacking algorithm. The heat leaking due to the lack of a heat trap occurs only during a non-draw cycle. Hot water in the second column of the water region of the water heater floats, through the connection between the pipe and the water tank, to the second (counting from the connection) node of the pipe model, and then cools off.

3. Draw Schedule

The energy consumption for supplying hot water is highly dependent on the daily draw schedule for the same total draw. A basic draw schedule (A) is taken from one particular sample of a field survey (13). Since there is neither identification of the purpose nor the flowrate of the draws, the flowrates of the chosen sample (Townhouse #5, day #88 of Ref. 13) are determined based on the time of the day of a draw and a suggested usage/flowrate relationship (14). Draw schedule A is given in Table 2.

In Table 2, the starting time of a draw is given in column 1, the flow rate in column 2, the duration of a draw in column 3, the volume of the draw in column 4 and the probable purpose in column 5. The durations of the draws are in units of hundredths of hours to conform with the time step of the water heater model.

An alternate draw schedule (B) is used to demonstrate the capability of the program to handle substantially different draw schedules. This alternate draw schedule is given in Table 3. It allows the same number of draws and same amount of total draw in a 24 hour period as Schedule (A). The prominent feature of this draw schedule is its two concentrated peaks (thus 2-peak schedule), one in the morning between 6 am and 9 am, the second one in the evening, between 6 pm and 10 pm.

4. Quasi-Steady State System Modeling

For the current parametric studies as described below quasi-steady state operation of the system for a typical 24-hour period was used. The assumption was made that the system is operated the same way day after day. In order to achieve this quasi-steady state, the system is modeled to operate more than a 24-hour period. The results then are checked for repeatability. If the results obtained at a time are nearly the same as that obtained 24 hours earlier, a quasi-steady state solution is achieved.

E. Summary of Parametric Studies

Case A. Basic System (Water Heater A, Pipe A). The basic system consists of the basic water heater without a heat trap, the basic pipe and draw schedule A.

Case B. Insulated Water Heater, Bare Pipe (WH B, Pipe A) The water heater is without a heat trap. Draw schedule A is used.

Case C. Insulated Water Heater, Insulated Pipe (WH B, Pipe B).

Case A-1. Basic System, Schedule (B). Same as Case A, draw schedule B is used.

Case A-2. Basic System with a Heat Trap. Same as Case A, the water heater having a heat trap.

Case A-3. Basic System, Schedule B, Reduced Thermostat Settings. Same as Case A-1 except the thermostat settings are reduced to maintain the same average faucet temperature for dishwashing draws.

V. Results and Discussions

A. Basic System (Case A)

The basic system consists of a water heater (WH A) with the thermostats set at 68.3°C (155°F) cutin and 71.1°C (160°F) cutout, and a distribution pipe (Pipe A). The system is operated on draw schedule A which is plotted in Figure 13(a). The flowrates are plotted versus time of the day. The numerals atop the spikes are durations of the individual draws, in hundredths of hours. The three draws starting at 20.00 hour have been identified as dishwashing draws. More details about the dishwashing draws are discussed later.

The thermal energy content of the water in the water heater referenced to 21.1°C (70°F) is plotted in Figure 13(b). It decreases exponentially during cooldown periods, and increases in a complex function—a combination of exponential terms during the recovery periods. During the periods of draws, the thermal energy content of water decreases in an approximately linear fashion. If during a period of draw the heating element is concurrently on, then of course, the thermal energy content of water decreases less rapidly (or may even increase).

The power input to the water heater is plotted in Figure 13(c). Since only one heating element may be operating at one time and each of the heating elements is rated at 4.5 kW, the power input is 4.5 kW anytime the power is on. The area under the power input curve is the total energy consumption of the water heater.

The average faucet temperature of the individual draws are plotted in Figure 13(d). The average temperature of a draw is a complex function of many factors: 1) the flowrate and the duration of the draw, 2) the flowrate and the duration of the previous draw, 3) the time elapsed between the previous draw and the current draw, 4) the heat transfer properties of the distribution pipe, 5) the temperature of the water before it enters the pipe, and 6) the ambient temperature. The factors (2) and (3) above give the initial condition for the pipe flow equations. The factors (5) and (6) give the boundary conditions for the pipe flow equations.

The three identical draws at .16 L/sec (2.5 gpm) starting at the 20.00 hour mark and lasting for .04 hours have been identified as probable dishwasher draws. The first draw achieves an average faucet temperature of 56.7C (134°F), the second 65°C (149°F), the third 67.2°C (153°F). The series of three identical draws at 12.00, 13.00 and 14.00 hour form another interesting case. The first one achieves an average faucet temperature of 43.3°C (110°F), the second 30.6°C (87.0°F), the third 26.1°C (79.0°F). The rising or falling of the average faucet temperature for three consecutive draws depends on the temperatures of the pipe at the beginning of each of the draws (or the initial conditions) as previously discussed. This property becomes clearer when the variations of the faucet temperature are discussed.

During non-draw periods the thermal energy content of water in the distribution pipe decreases in an exponential fashion similar to water heater cooldown. Upon the start of a draw the thermal energy content of the pipe increases rapidly. Similarly, the rate of heat loss from the pipe decreases in an exponential fashion during non draw periods since it is directly related to the thermal energy content of the water in the pipe. This is demonstrated in Figure 13(e) at the 8.40 hour mark where after achieving a peak due to three heavy draws, the rate of heat loss starts to decrease exponentially until the onset of the next draw. At that time, it increases rapidly to a high value.

The rates of heat loss from the water heater are plotted against time of the day in Figure 14. The rate of heat loss through the top surface remains almost constant throughout the day. This is because the water in the upper portion of the tank remains approximately 71.1°C (160°F) due to the stratification which is modeled. The cold supply water is delivered to the bottom of the tank. If the stored water heat content were substantially depleted, the water temperature in the upper portion of the tank would drop significantly below 71.1°C (160°F). The fact that the hot water content is not substantially depleted may be demonstrated by Figure 13(b) where the water energy at the lowest point is still 68% of the fully recovered value. The rates of heat loss of the lateral surfaces and of the bottom, on the other hand, are sensitive to the draws as shown in Figure 14. Note the synchronous variation of the rates of heat loss of the lateral surfaces and the bottom, and the water energy curves.

The faucet temperature variations of the selected draws are plotted in Figure 15. A typical faucet temperature variation is that of the 7.00 hour draw. Starting at 28.7C (83.7°F), it dips slightly and momentarily before increases rapidly in the first .01 hour. Its rate of increase gradually reduces until at large time the faucet temperature reaches a steady-state value before the draw terminates. For shorter and smaller draws, the faucet temperature for the 18.00 hour draw is typical. Starting at 25.9°C (78.7°F), it reduces to 25.6°C (78°F) .009 hour later. It then increases to 27.3C (81.2°F) at .02 hour after the start of the draw. The reduction of the faucet temperature right after the onset of a draw is a characteristic of the distribution pipe with a vertical section at the faucet end of the pipe. Because of the vertical section, a portion of its water temperature is warmer than the rest of the pipe due to the buoyancy effect. When the faucet is opened, the warmer water flows out first. After the segment of warmer water is exhausted the colder water flows out of the faucet, before the hot water arrives from the water heater.

The falling of the average faucet temperatues of the three draws at 12:00, 13:00 and 14:00 hour and the rising of the average faucet temperatures of the three dishwashing draws can now be clarified. The former three draws have decreasing initial temperatures while the latter three draws have increasing initial temperatures.

Table 4 summarizes for all cases studied, the electrical energy consumption (Electricity), the thermal energy in the delivered water relative to the supply water (Hot Water), the total heat loss (Loss) for a 24-hour period, and the net difference in the energy balance over the 24-hour period (Convergence). The electrical energy consumption is that consumed by the electrical heating elements and the hot water delivery is the energy delivered at the faucet.

The total heat loss consists of the heat loss through the jacket of the water heater (top, bottom, and lateral surfaces), and the walls of the distribution pipe. The convergence is the change of the total energy of the system in a 24-hour period. The same energy consumptions are presented in Table

5 as the percentages of the electrical energy consumptions. The total heat loss, as a percentage of the electrical energy consumption, indicates the inefficiency of the operation of a given system.

1. Schedule B (Case A-1)

The basic (Case A) model is used to study of the effect of changes in draw schedule. The same total volume draw of 438.5 L (115.8 gallon), is delivered by the system, the only difference being the time schedule of the draws. The differences in energy supplied, losses, and in hot water between the two draw schedules are shown in Table 4.

The differences in the efficiency with a concentrated draw schedule may be slightly less pronounced in a multi-branching pipe than in the single pipe network. This is because in a single-pipe network, the entire length of pipe contributes to changing the hot water delivery and reducing the pipe heat loss when a concentrated draw schedule is used. In a multi-branching pipe network, however, only the main line of the network contributes to the change when a concentrated draw schedule is used.

2. Effects of a Heat Trap (Case A-2)

A distribution pipe is usually connected to a water heater with a vertical section. The water in the vertical section cools off much faster than the water in the water heater, because 1) the pipe section is usually not insulated that 2) the pipe has a surface-to-volume ratio which is much greater than that of the water heater. The effect of the heat trap is studied when it is modeled on the Case A system.

If there is no heat trap installed at the pipe to water heater connection, the colder water in the vertical pipe section sinks to the water heater. The same amount of hot water rises from the water heater to the pipe section to take the place of the colder water. The hot water again loses heat and be replaced by hot water from the water heater. This thermal syphon process is taking place continuously during the non-draw periods. An ideal heat trap stops this process.

The effectiveness of a heat trap is demonstrated in Table 4 when comparing the results of Case A-2 with those of Case A. The heat loss is reduced by 3.06 MJ (2900 Btu). In terms of the distribution of the energy consumptions (Table 5), the heat loss is reduced from 25.8% of the electricity consumption to 23.7%.

B. Insulated Water Heater (Case B)

The rates of heat loss from the insulated water heater are plotted in Figure 16. As might be expected the rates of heat loss of the lateral and the bottom are synchronous with the thermal energy of the water in the heaters in

Case A. The values of these rates of heat loss for Case B are much lower than the corresponding values for Case A except for the rate of heat loss of the bottom which is not affected (no insulation was added to the bottom of the tank). The insulating of the water heater causes the heat loss to reduce from 29.01 MJ (27500 Btu) to 26.27 MJ (24900 Btu), and the daily electricity consumption to reduce from 31.28 kWh to 30.64 kWh. In terms of electricity consumption, the heat loss decreases from 25.8% to 23.8%.

The reduction in heat loss is different from that of a simpler calculation discussed in section II, because the simpler calculation does not account for: the heat loss of the distribution pipe, the heat capacity of the insulating material, the temperature dependence of the overall conductance and the temperature stratification in the tank.

C. Pipe Insulation (Case C)

Tables 4 and 5 show that when the pipe is insulated in addition to the water heater the heat loss is reduced substantially. For this example, losses were reduced from 23.8% to 20% of the electrical energy used in a 24 hour period.

D. Thermostat Setback (Case A-3)

The effects of a 5.6°C (10°F) thermostat setback are also shown in Tables 4 and 5. For this example, the energy required to meet the requirements of draw schedule A are reduced from 31.36 to 28.76 kWh and the losses are reduced proportionally.

E. Water Delivery Temperature (Faucet Temperature)

To illustrate the capability of the model to predict the changes in water delivery temperature with time, the water delivery temperatures for three successive identical 22.7 L (6 gal.) draws were computed, Figure 17, with and without pipe insulation (Cases C and B, respectively). The first draw starts at 18.00 hour mark, and lasts until 18.04 hour mark; the second draw starts at 18.25 and lasts until 18.29 hour mark; the third draw starts at 18.33 and lasts until 18.37 hour mark. Thus, the three draws are separated by two cooldown periods. For this example, it is noted that the insulated pipe maintains higher delivery temperature than that of the bare pipe during the entire period of the first draw. In the second draw, however, the bare pipe gives higher delivery temperature than the insulated pipe initially. Later on, the insulated pipe again delivers hotter water.

The reason: subsequent to the termination of the first draw, in case of the bare pipe, the water is losing heat via the pipe wall to the ambient, in case of the insulated pipe, however, the water is losing heat, via the pipe, to the insulation which, having a thermal diffusivity 1000 times greater than that of copper, is still relatively cold. In case of the bare pipe, the convective heat transfer between the pipe and the ambient is the controlling factor determining the rate of heat loss, in case of insulated pipe, the conduction heat transfer between the pipe and the insulation is the controlling factor determining the rate of heat loss. The former heat transfer is smaller than the latter heat transfer during the first cooldown period. The net result: the bare pipe has hotter water than the insulated pipe at the start of the second draw. A similar situation occurs for the third draw. If a cooldown period lasts for a long time giving enough time for the pipe insulation to heat up (so insulation is no more a heat sink), then the overall thermal resistance between the water and the ambient, now the determining factor, is higher in the insulated pipe than that in the bare pipe. The net result: the insulated pipe has hotter water than a bare pipe, e.g., the first draw.

VI. Conclusions

A computer model has been developed to study the dynamic performance of the hot water system of a residence and some commercial facilities.

The model may be used to study the effects of changing the following parameters on the performance of the system:

A. Pertaining to the water heater:

1) the size of tank,

2) the diameter-to-height ratio of the tank,

3) the location and size of the heating elements,

4) the location of the thermostats and their settings,

5) the property and thickness of the tank wall and, particularly the insulation.

B. Pertaining to distribution pipe:

1) the length and the size of the pipe,

2) the location of horizontal and vertical sections of the pipe,

3) the material of the pipe,

4) the insulation of the pipe.

C. <u>Pertaining to the System:</u>

1) the draw schedule,
2) the presence of a heat trap,
3) the location of the connection of the distribution pipe to the water heater,
4) the temperature of the supply water,
5) the ambient temperature.

A parametric study has been performed to illustrate quantitatively the effects of:

1) insulating the water heater,

2) insulating the distribution pipe,

3) changing the draw schedule,

4) installing a heat trap, and

5) thermostats setback.

It is recommended that an experimental study be performed to verify or refine the analytical dynamic model. A verified analytical model could form the basis of simpler models for use as design tools in predicting thermal performance of water heating systems.

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	Coefficients for Equations 41 and 42			Thermal Conductivity, $k, \frac{\text{watt}}{\text{mK}}$		
	C_o	C_t	C_b	Insulation	Extra Insulation	Copper
Water Heater A	5.5	2.6	4.5	.048	--	--
Water Heater B	5.5	2.6	4.5	.048	.046	--
Pipe A	See text	--	--	--	--	418
Pipe B	See text	--	--	--	.23	418

Characteristic Dimension of Room = 4.5 m

Emmissivity of Water Heater Jacket = 0.6

Emissivity of Walls of Room = 0.9

Emissivity of Pipe Surface = 0.6

Table 1 Selected Heat Transfer Parameters for Modeling

Time of Day (Hour)	Flowrate L/sec (GPM)	Duration (Hour)	Volume L(Gallon)	Purpose
0.00	0.032(0.5)	0.01	1.14(0.3)	
1.00	0.095(1.50)	0.03	10.2(2.7)	
2.00	0.032(0.50)	0.01	1.14(0.3)	
3.00	0.032(0.50)	0.08	9.09(2.40)	
4.00	0.032(0.50)	0.02	2.27(0.60)	
5.00	0.032(0.50)	0.02	2.27(0.60)	
6.00	0.032(0.50)	0.02	2.27(0.60)	
7.00	0.088(1.40)	0.23	73.1(19.32)	Bath
8.00	0.17(2.70)	0.05	30.7(8.10)	Clothes Washing
8.33	0.17(2.70)	0.05	30.7(8.10)	Clothes Washing
8.42	0.17(2.70)	0.05	30.7(8.10)	Clothes Washing
11.00	0.095(1.50)	0.03	10.2(2.70)	
12.00	0.032(0.50)	0.01	1.14(0.30)	
13.00	0.032(0.50)	0.01	1.14(0.30)	
14.00	0.032(0.50)	0.01	1.14(0.30)	
15.00	0.095(1.50)	0.08	27.30(7.20)	
17.00	0.032(0.50)	0,01	1.14(0.30)	
18.00	0.032(0.50)	0.02	2.27(0.60)	
20.00	0.16(2.50)	0.04	22.7(6.00)	Dish Washing
20.25	0.16(2.50)	0.04	22.7(6.00)	Dish Washing
20.33	0.16(2.50)	0.04	22.7(6.00)	Dish Washing
21.00	0.088(1.40)	0.11	35.0(9.24)	Shower
22.00	0.088(1.40)	0.11	35.0(9.24)	Shower
23.00	0.16(2.50)	0.11	62.5(16.50)	Bath

Table 2 Draw Schedule A

Time of Day (Hour)	Flowrate L/Sec(GPM)	Duration (Hour)	Draw Volume L (Gallon)	Purpose
6.80	0.032(0.50)	0.01	1.14(0.3)	
6.90	0.032(0.5)	0.01	1.14(0.3)	
7.00	0.088(1.40)	0.23	73.1(19.32)	Bath
7.50	0.17(2.70)	0.05	30.7(8.10)	Clothes Washing
7.83	0.17(2.70)	0.05	30.7(8.10)	Clothes Washing
7.92	0.17(2.70)	0.05	30.7(8.10)	Clothes Washing
8.00	0.095(1.50)	0.03	10.2(2.70)	
8.10	0.032(0.5)	0.01	1.14(0.30)	
18.50	0.095(1.50)	0.08	27.3(7.20)	
18.60	0.032(0.50)	0.01	1.14(0.30)	
18.70	0.032(0.50)	0.02	2.27(0.60)	
19.00	0.088(1.40)	0.11	35.0(9.24)	Shower
19.50	0.088(1.40)	0.11	35.0(9.24)	Shower
19.70	0.095(1.50)	0.03	10.2(2.70)	
19.80	0.032(0.50)	0.08	9.09(2.40)	
20.00	0.16(2.50)	0.11	62.5(16.50)	Bath
20.50	0.16(2.50)	0.04	22.7(6.00)	Dishwashing
20.75	0.16(2.50)	0.04	22.7(6.00)	Dishwashing
20.83	0.16(2.50)	0.04	22.7(6.00)	Dishwashing
21.00	0.032(0.50)	0.01	1.14(0.30)	
21.10	0.032(0.50)	0.01	1.14(0.30)	
21.20	0.032(0.50)	0.02	2.27(0.60)	
21.30	0.032(0.50)	0.02	2.27(0.60)	
21.40	0.032(0.50)	0.02	2.27(0.60)	

Table 3 Schedule B (Two-Peak Schedule)

CASES

	A	A-1	A-2	A-3	B	C
Electricity	112.68	113.00	109.62	103.62	110.36	109.73
Hot Water	83.62	89.53	83.51	82.46	83.93	87.31
Loss	29.01	22.68	25.95	20.47	26.27	21.95
Convergence	0.05	0.79	0.16	0.69	0.16	0.47

Case A - basic WH, basic pipe, Schedule A, no trap
Case B - insulated WH, basic pipe, Schedule A, no trap
Case C - insulated WH, insulated pipe, Schedule A, no trap
Case A-1 - Case A with Schedule B
Case A-2 - Case A with heat trap
Case A-3 - Case A-1 with 10°F WH thermostat setback

Table 4 Energy Consumptions (in MJ)

	CASES					
	A	A-1	A-2	A-3	B	C
Electricity	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%
Hot Water	74.2%	79.2%	76.2%	79.5%	76.0%	79.6%
Loss	25.8%	20.0%	23.7%	19.8%	23.8%	20.0%
Convergence	0.0%	0.8%	0.1%	0.7%	0.2%	0.4%

Table 5 Distribution of Energy Consumption

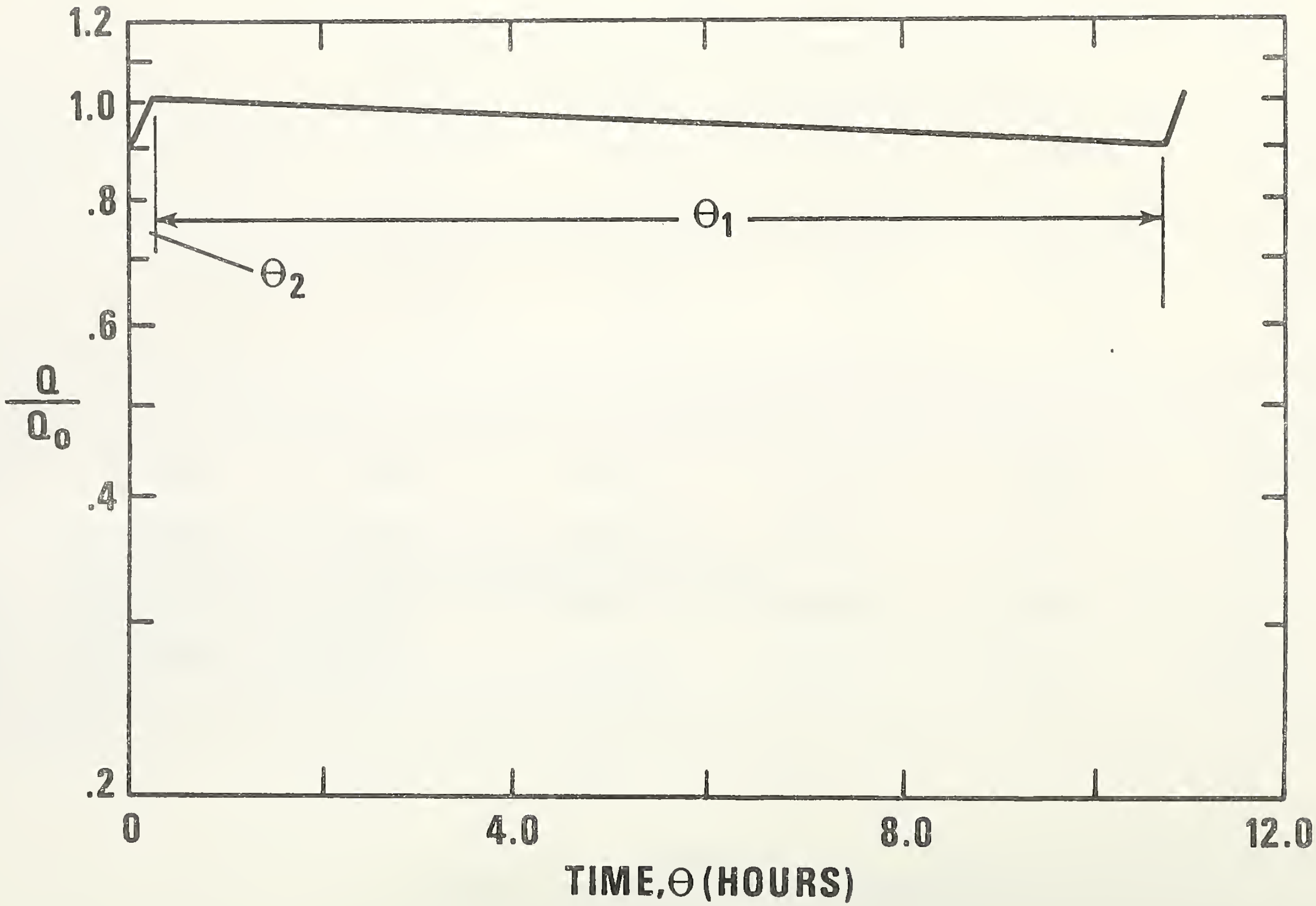


Figure 1   Bulk standby of an electric water heater.

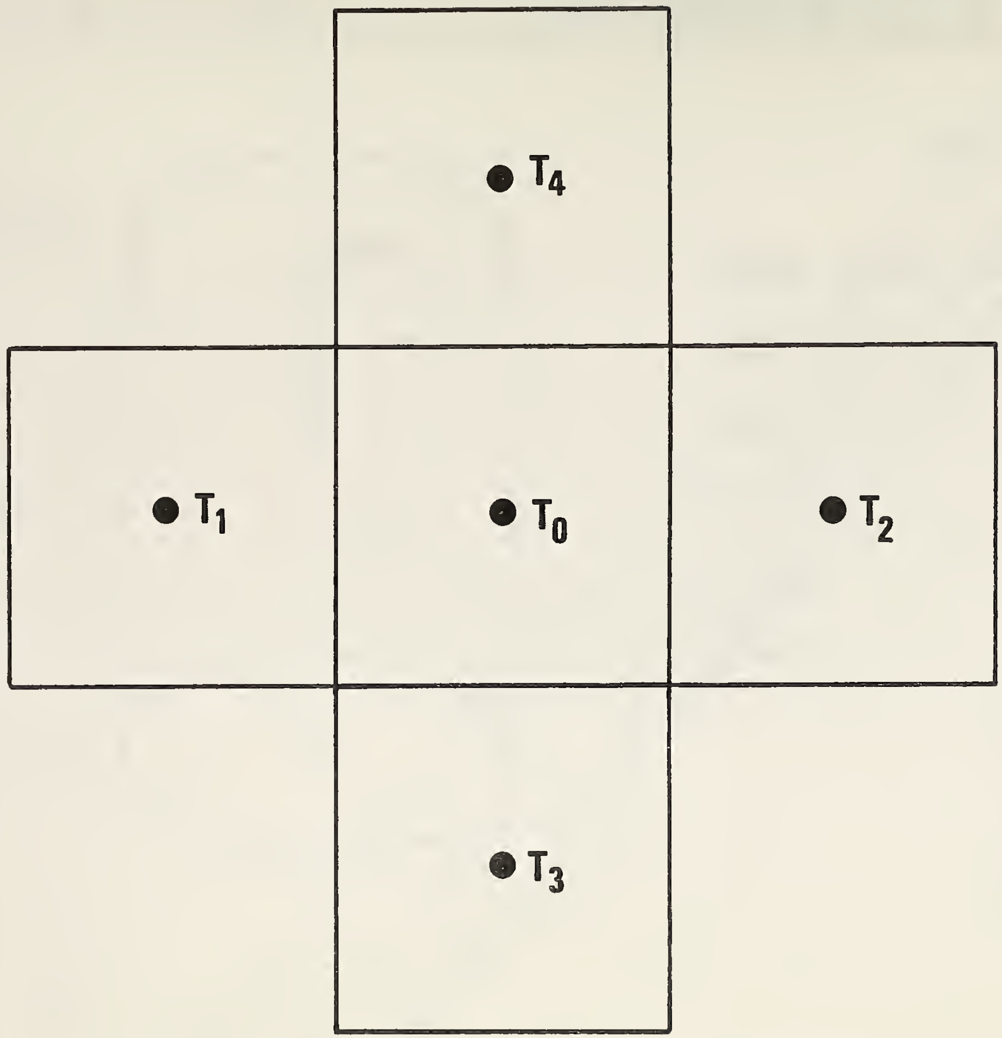


Figure 2 Nodal scheme for the conduction model.

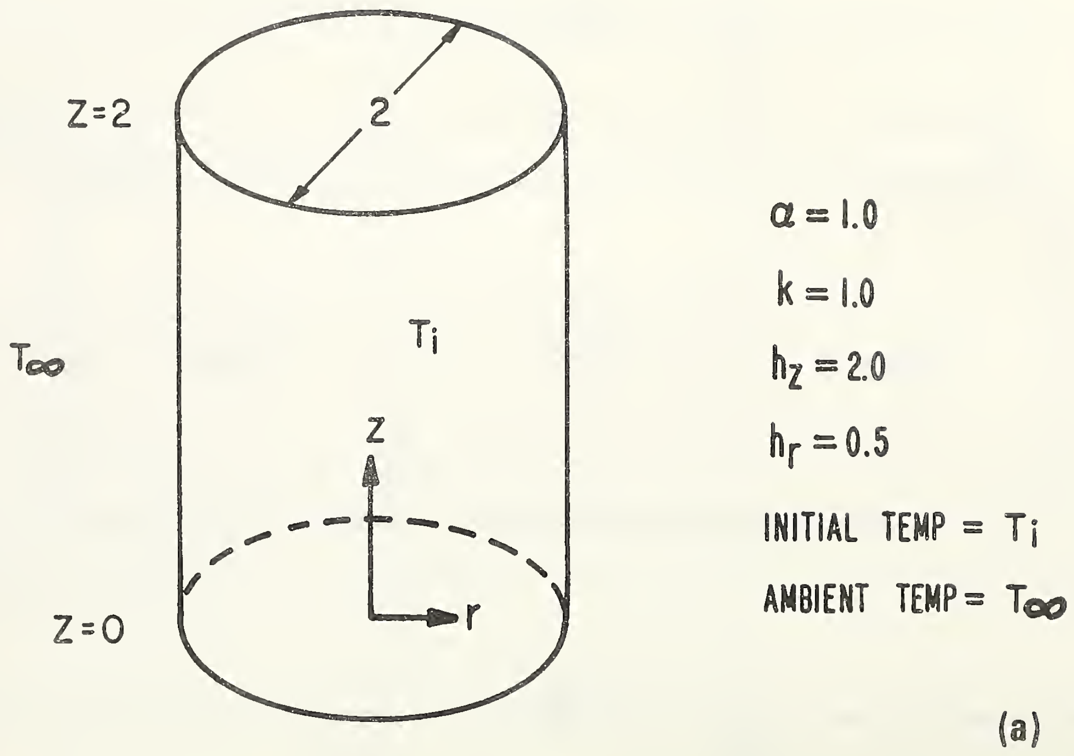


Figure 3 Verification of the conduction model of ESDS.

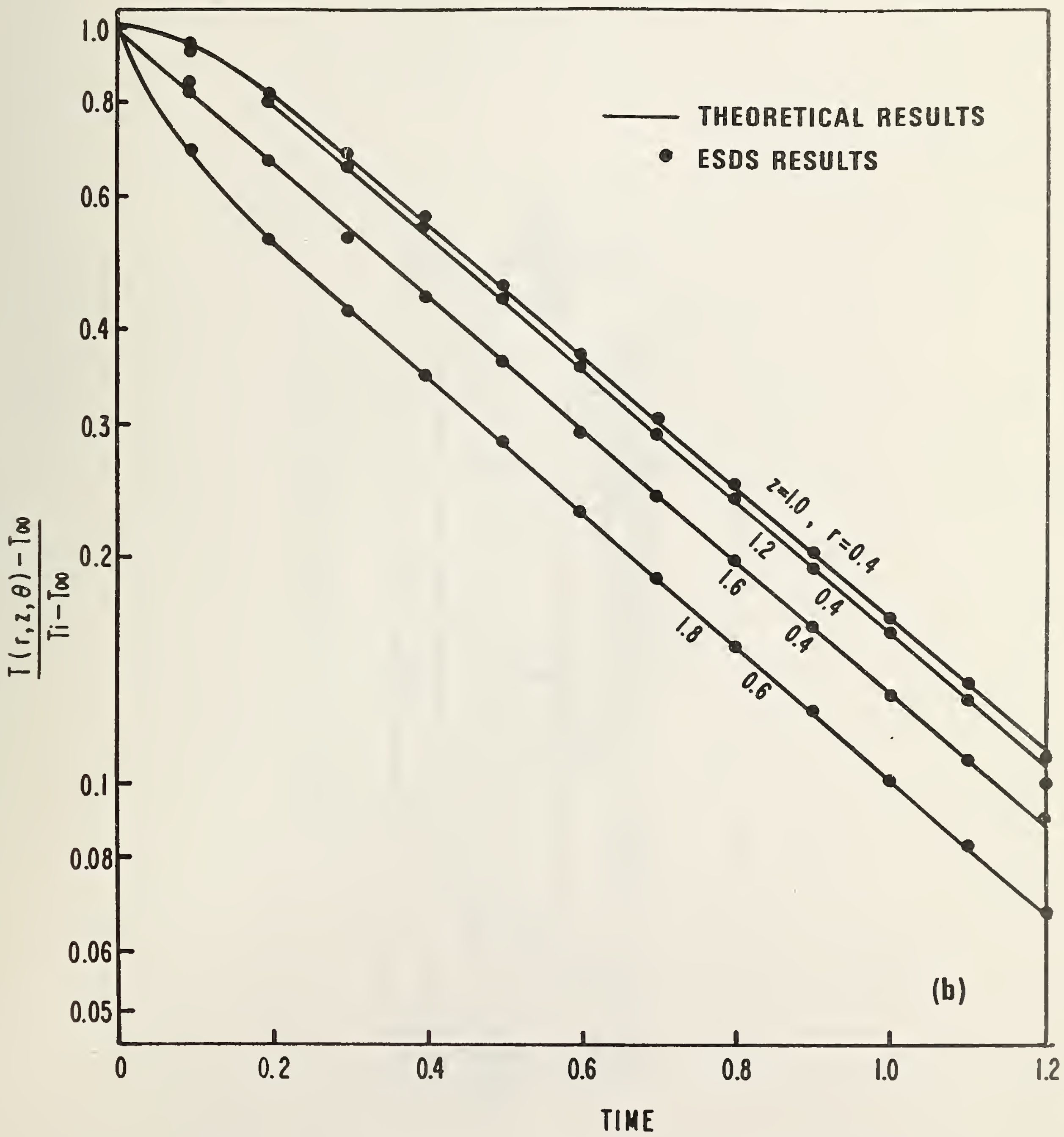


Figure 3 Verification of the conduction model of ESDS.

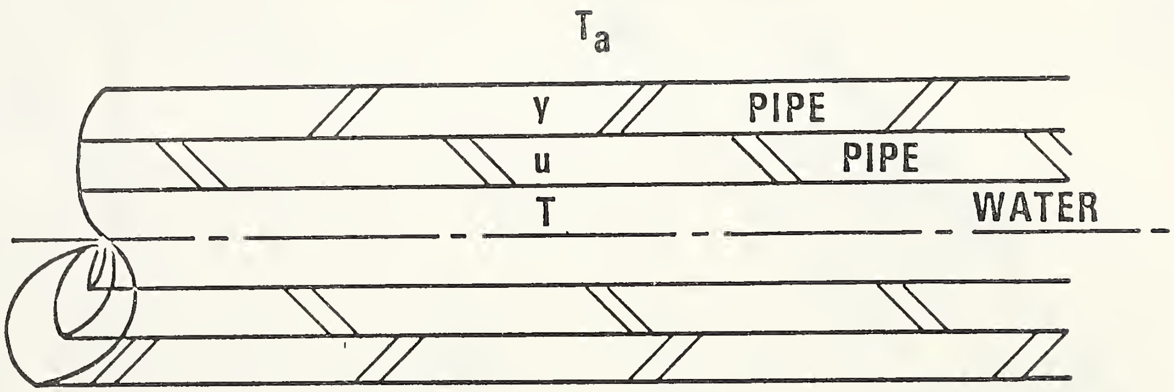


Figure 4   Two layer pipe model.

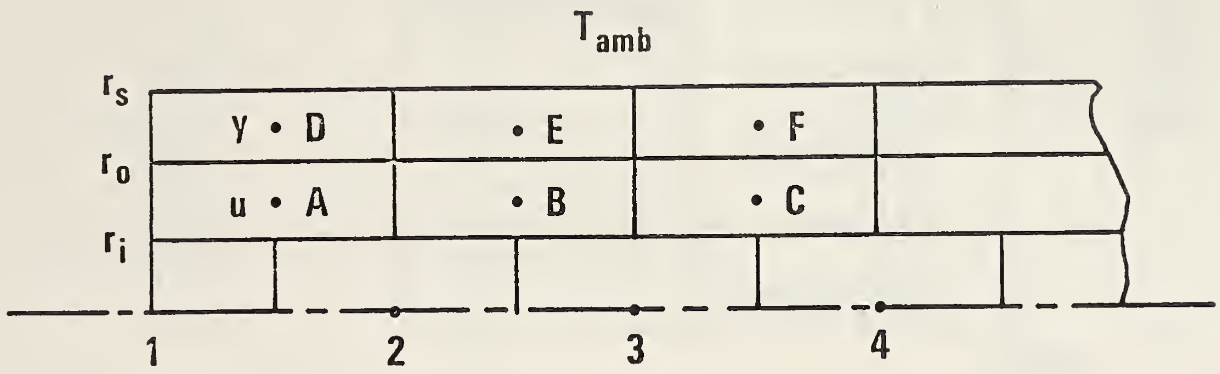


Figure 5   Nodal scheme for the pipe flow model.

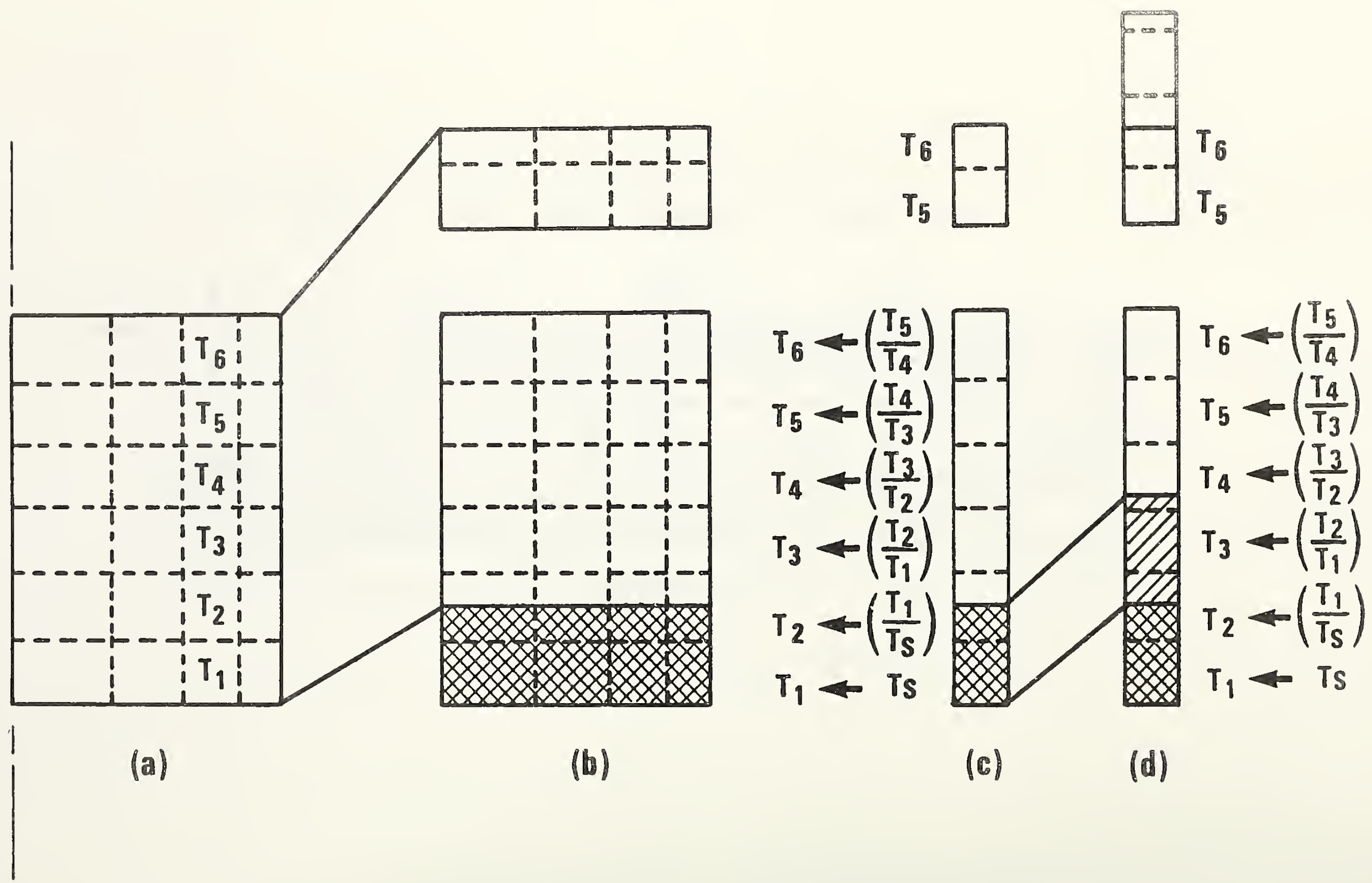


Figure 6   Draw model, Option 1.

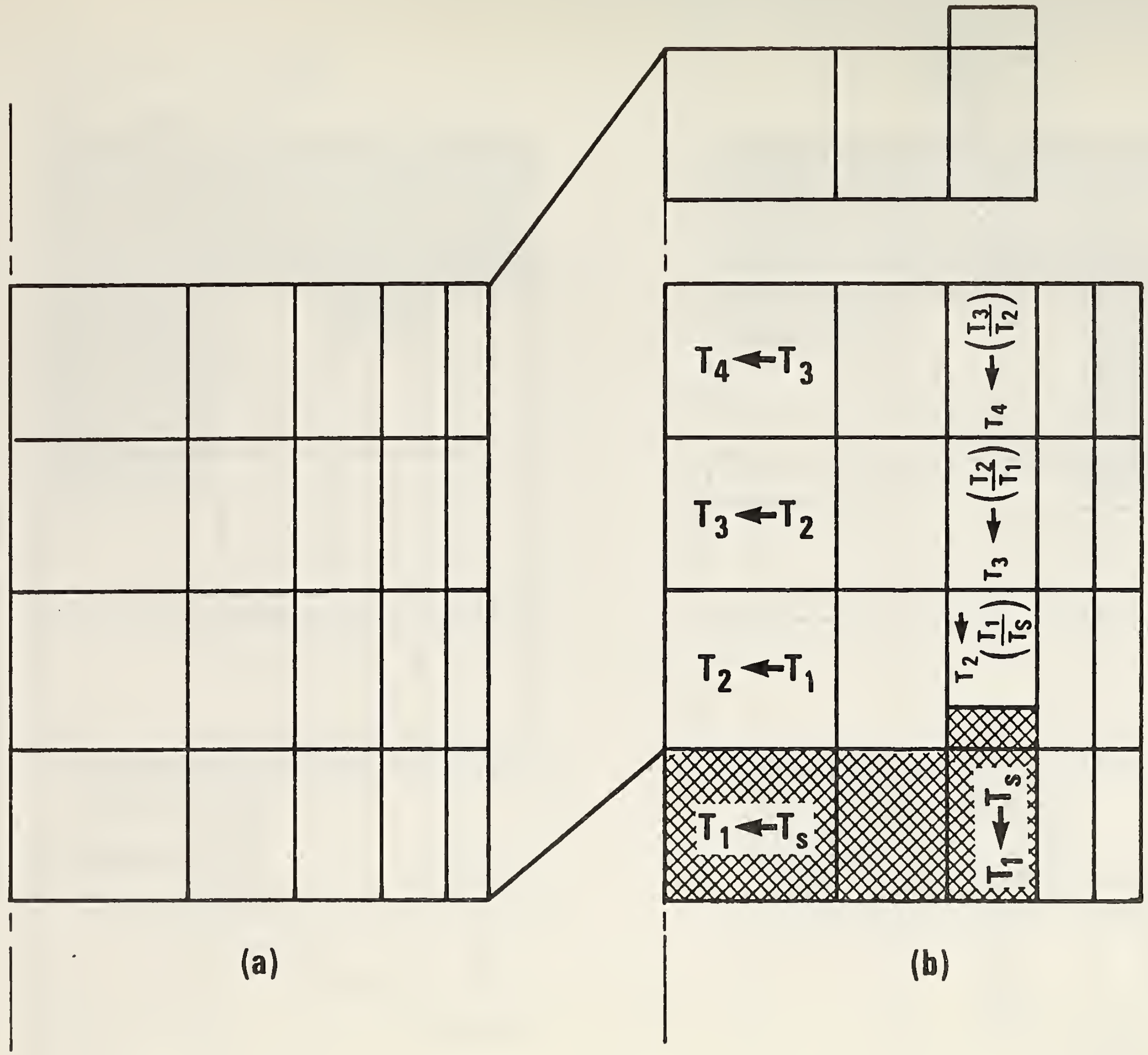


Figure 7 Draw model, Option 2.

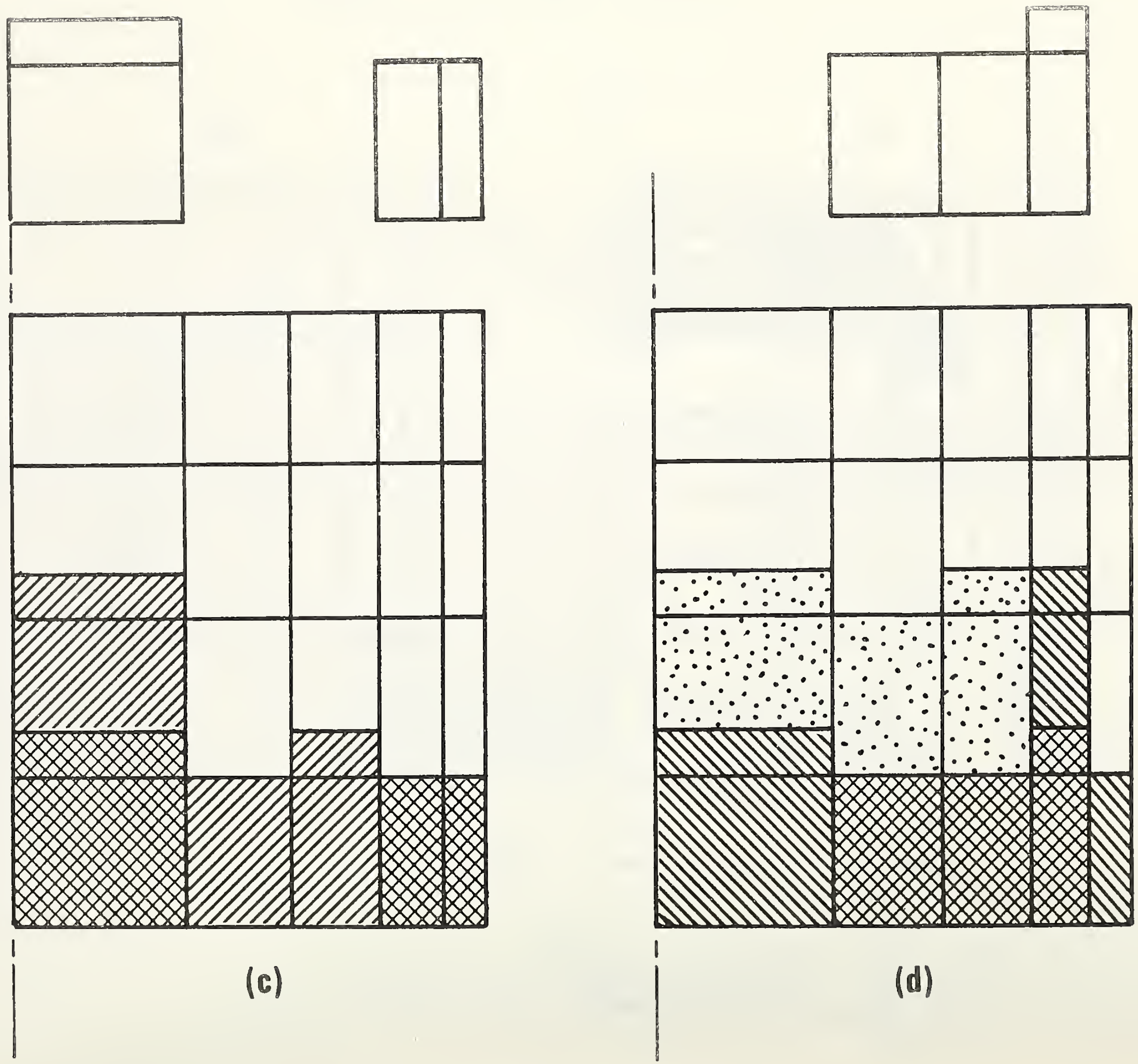


Figure 7 Draw model, Option 2.

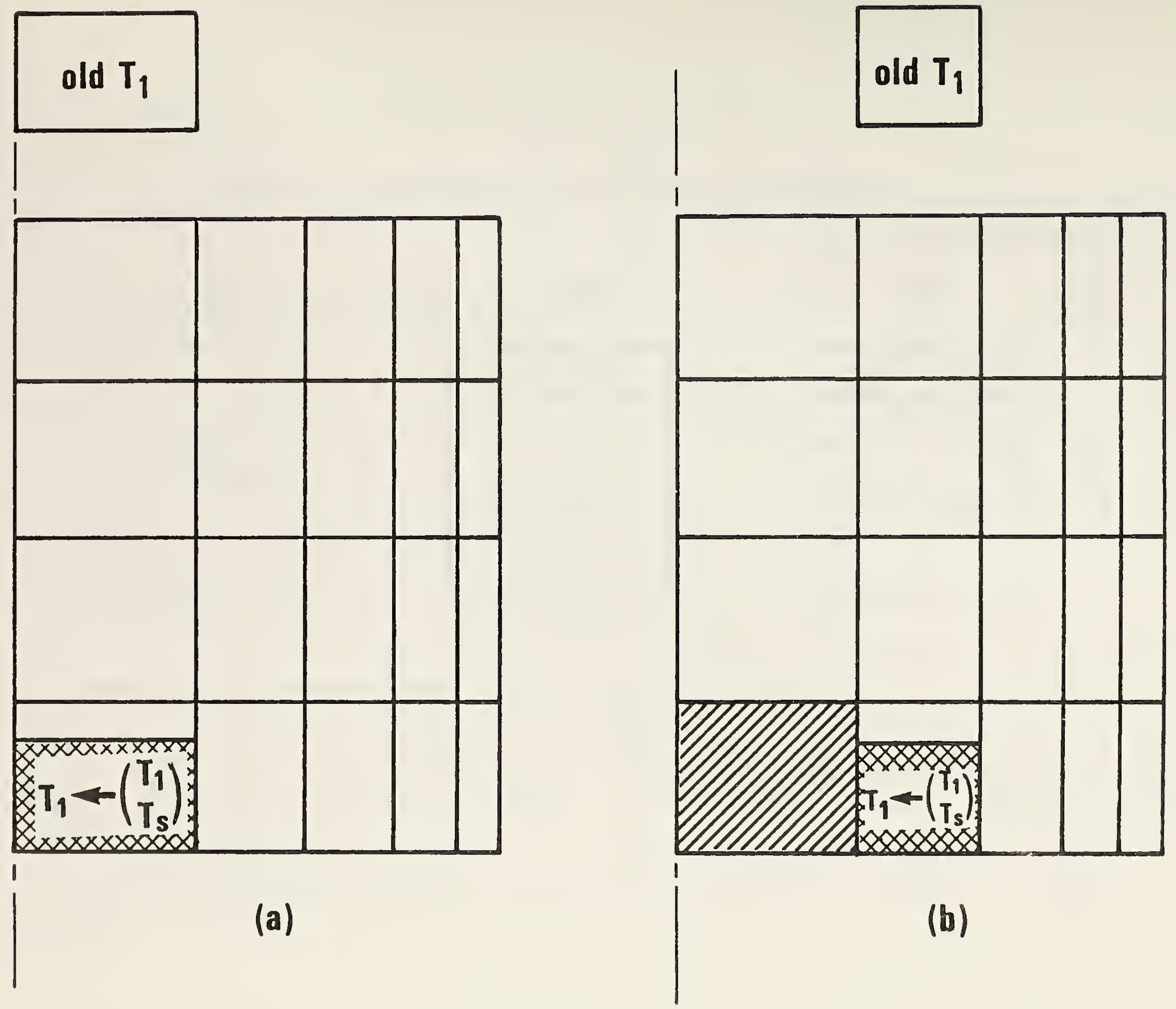


Figure 8 Draw model, Option 3.

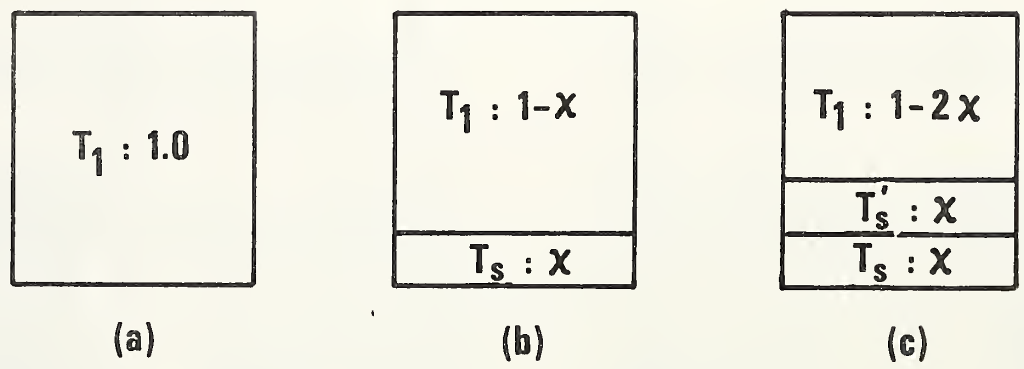


Figure 9 Water node temperature for small draw.

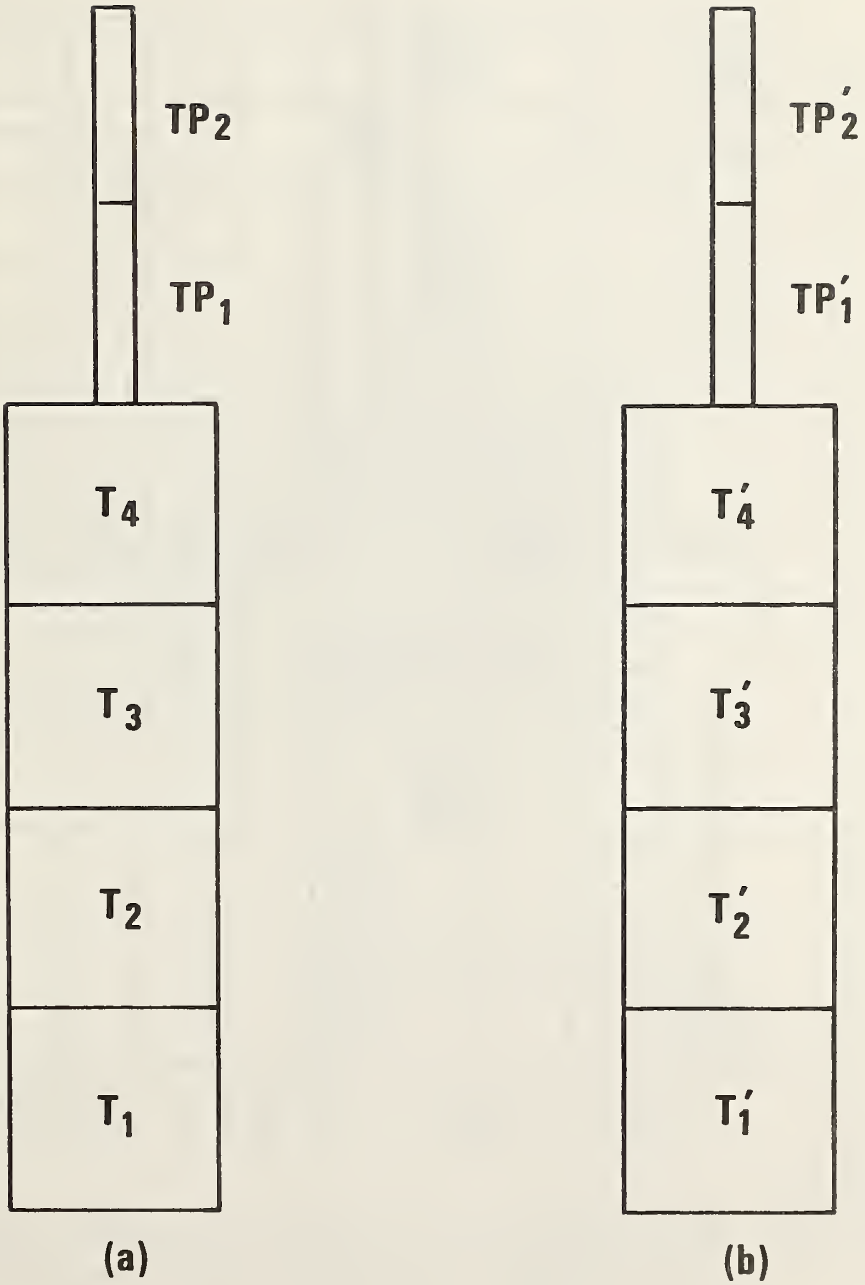


Figure 10 Heat leak through a stand pipe.

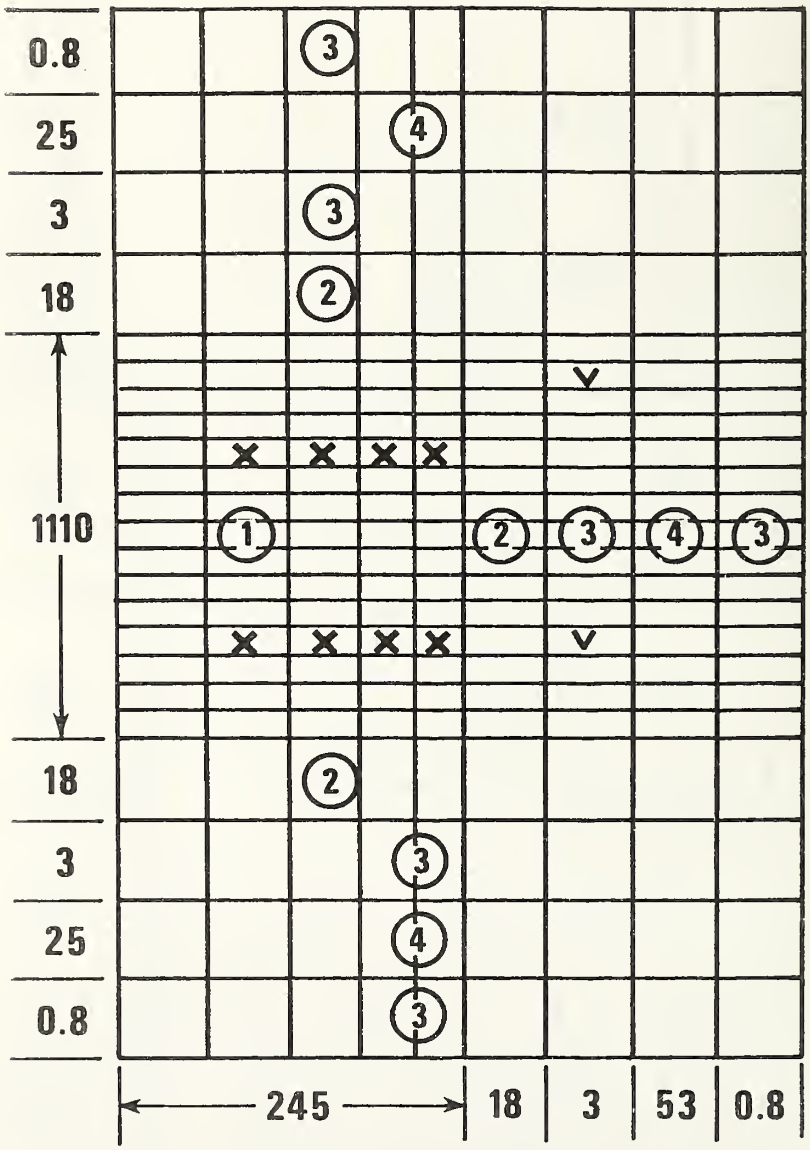
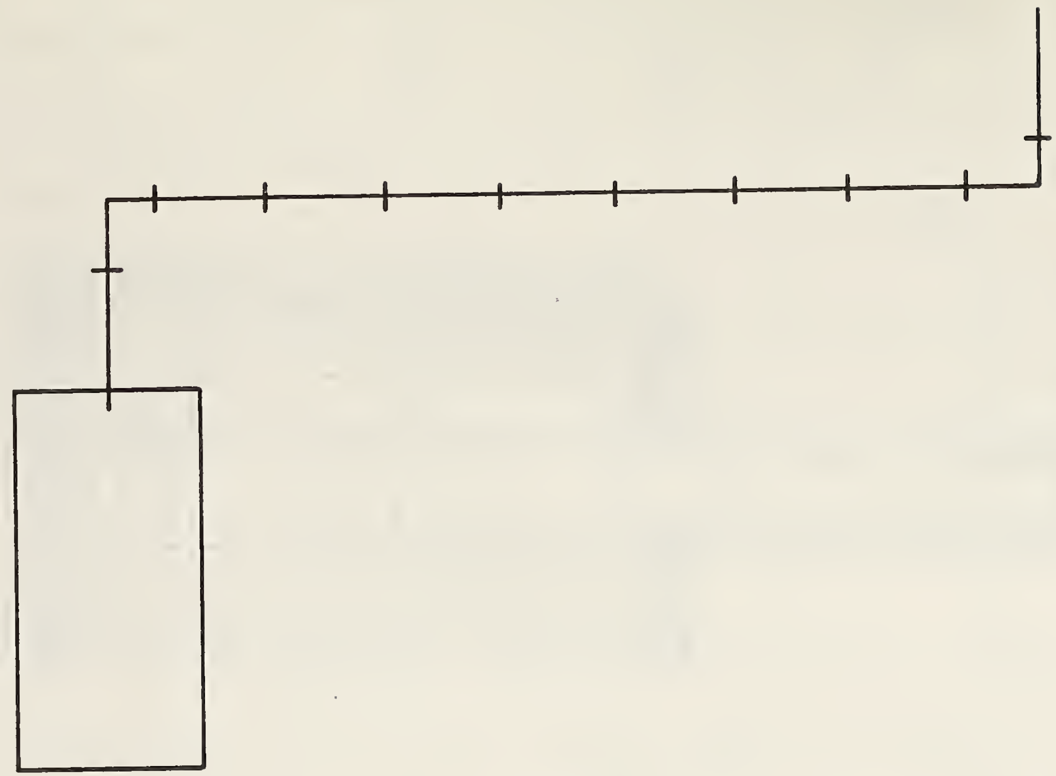
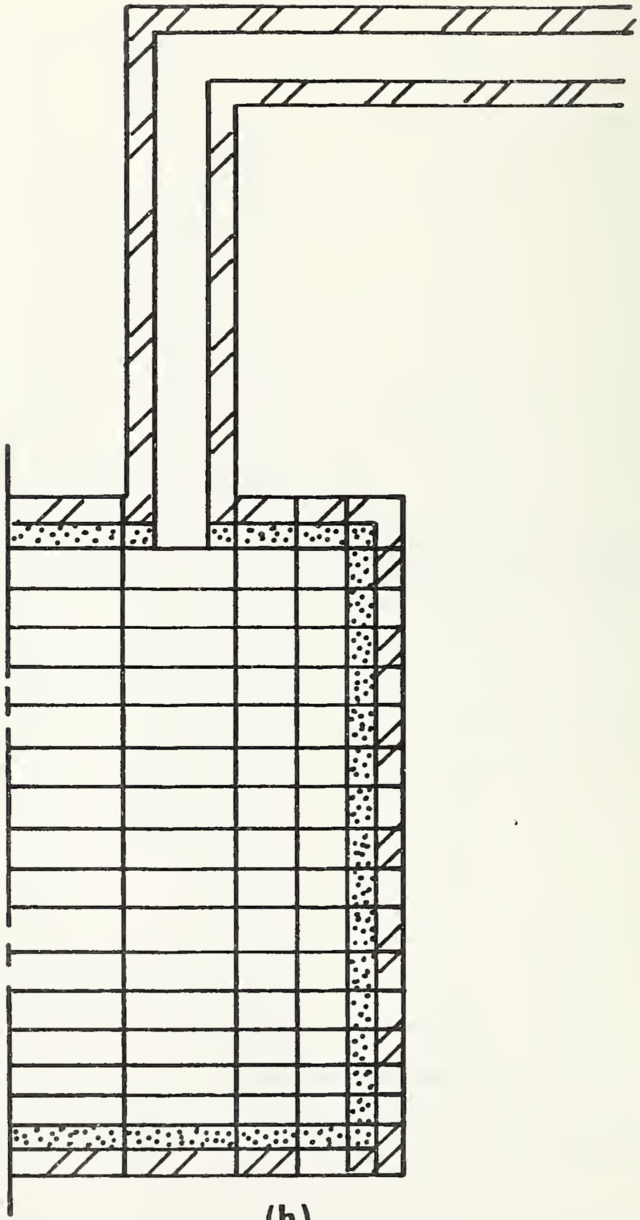


Figure 11 Nodal diagram for the basic water heater (water heater A) Dimensions in mm. (1) water, (2) cement, (3) steel, (4) insulation. X heating source, V thermostat.



(a)

Figure 12 Nodal diagram for the pipe and its connection to the water tank.



(b)

Figure 12 Nodal diagram for the pipe and its connection to the water tank.

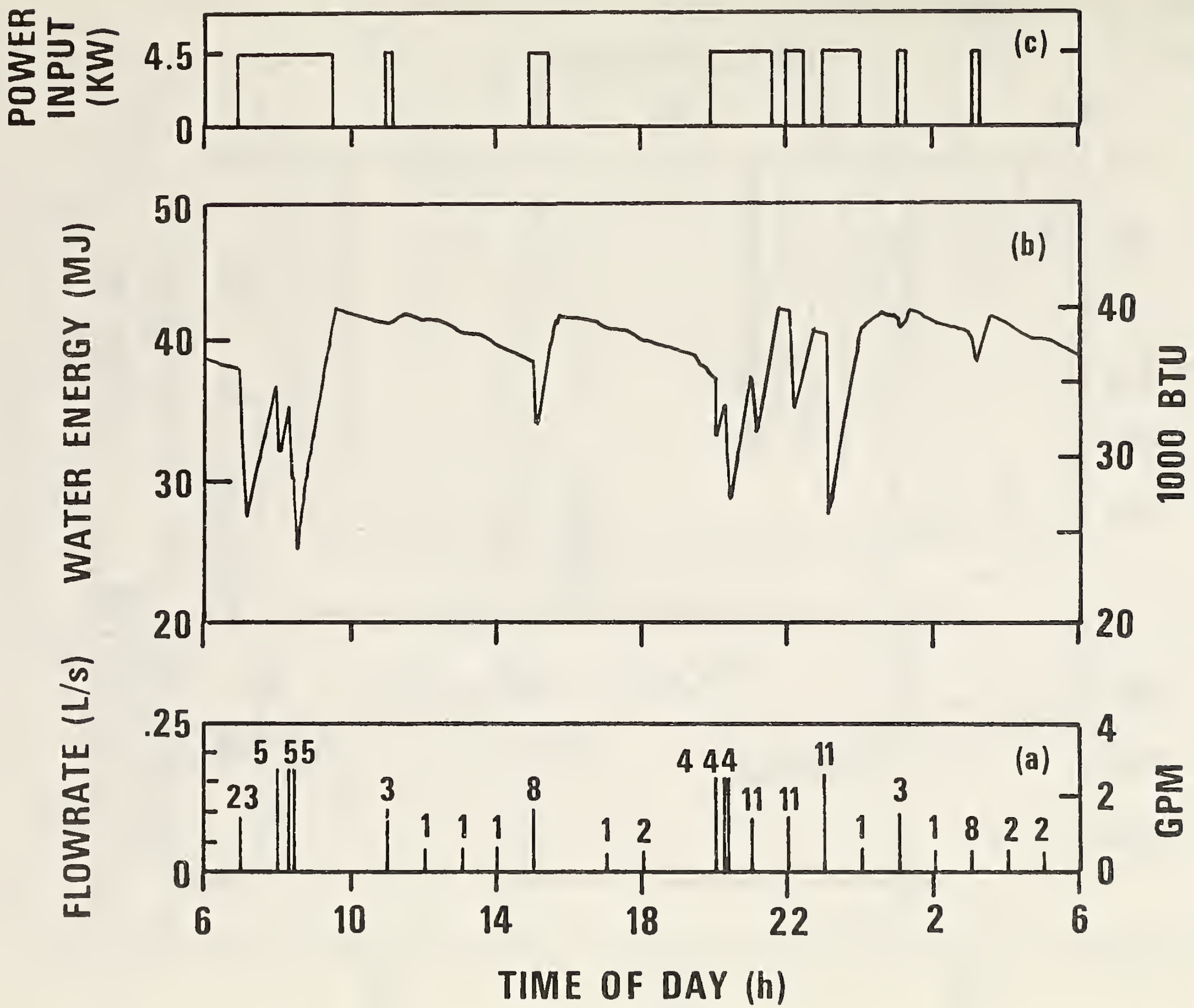


Figure 13 A 24-hour operation of Case A, in which Draw Schedule A is used.

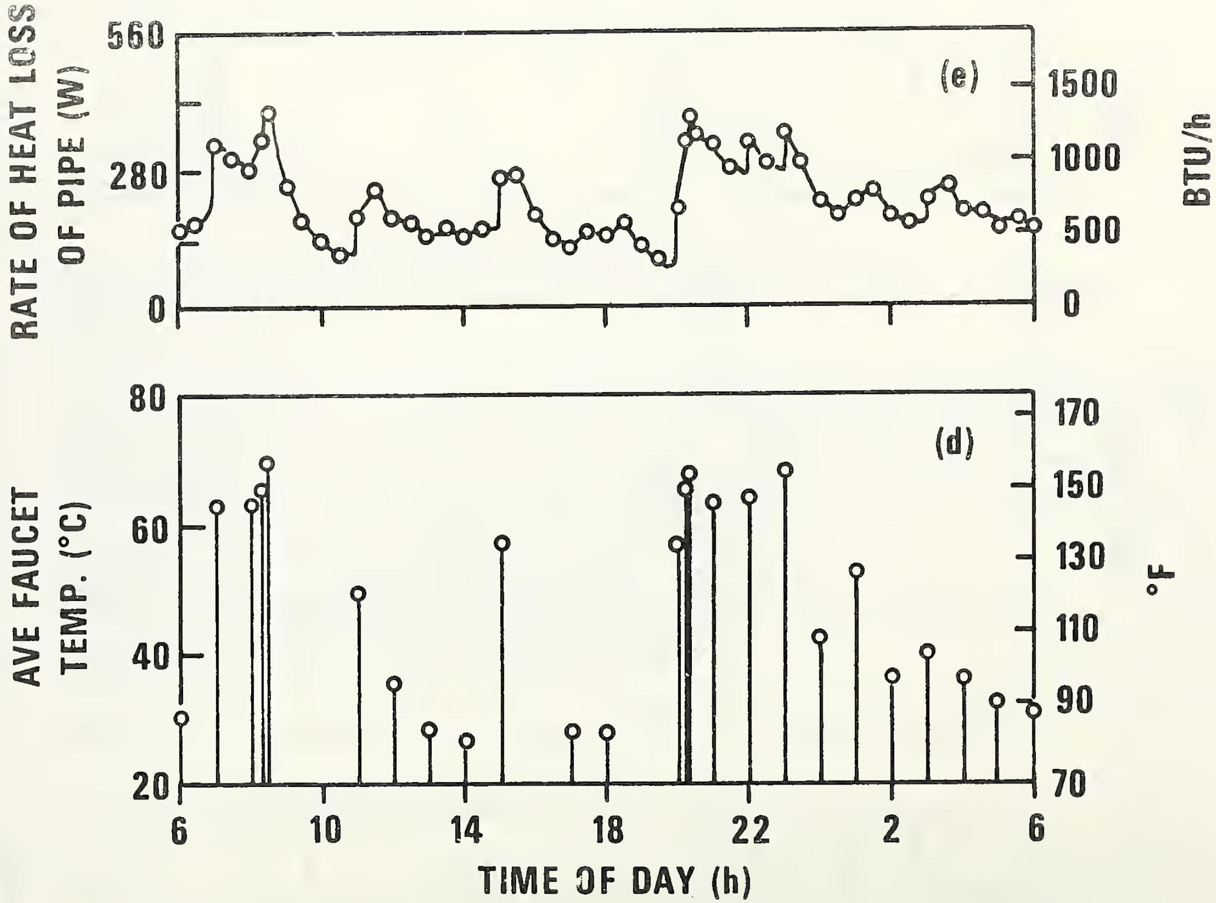


Figure 13 A 24-hour operation of Case A in which Draw Schedule A is used.

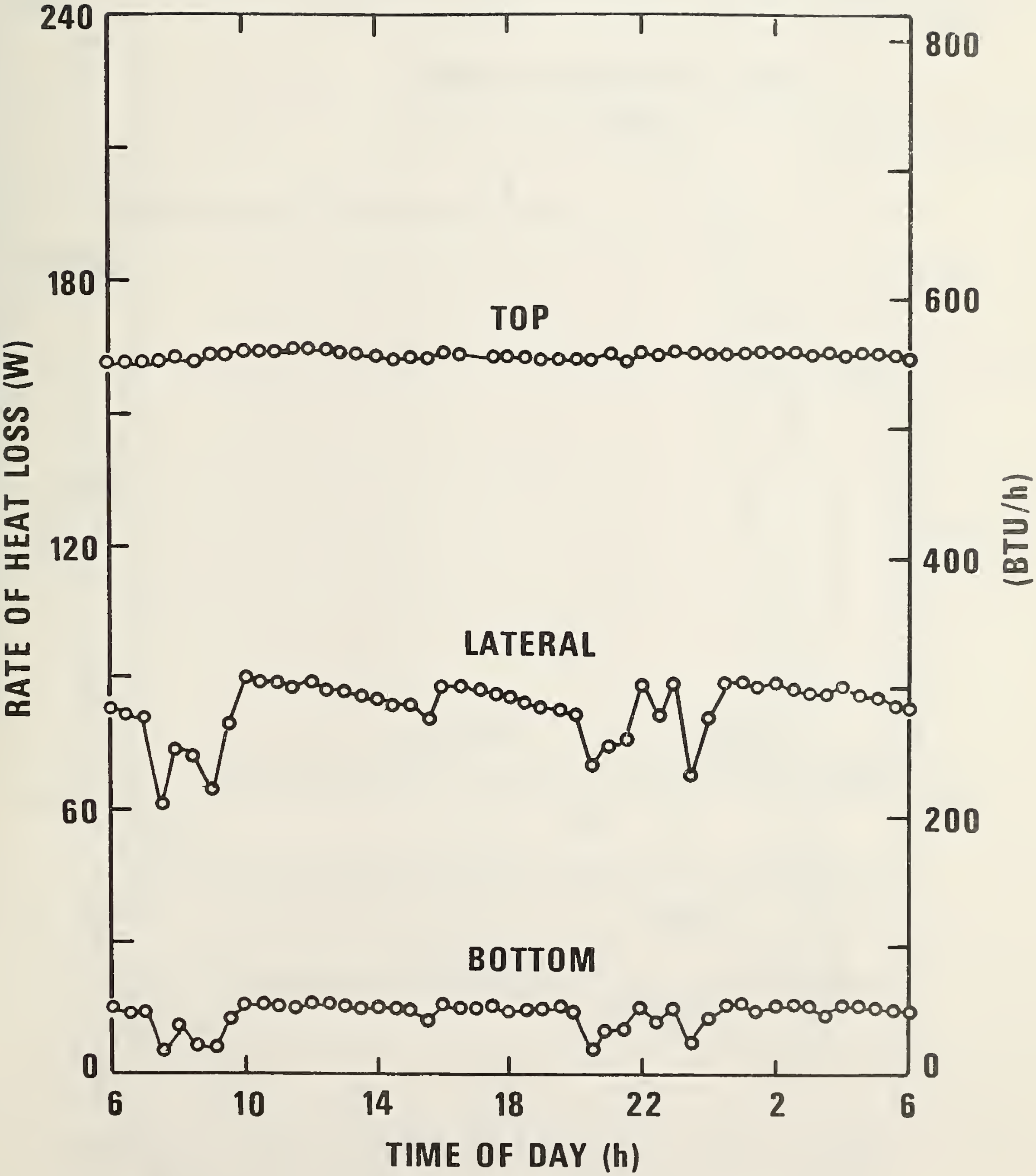


Figure 14 The rates of heat loss of water heater A.

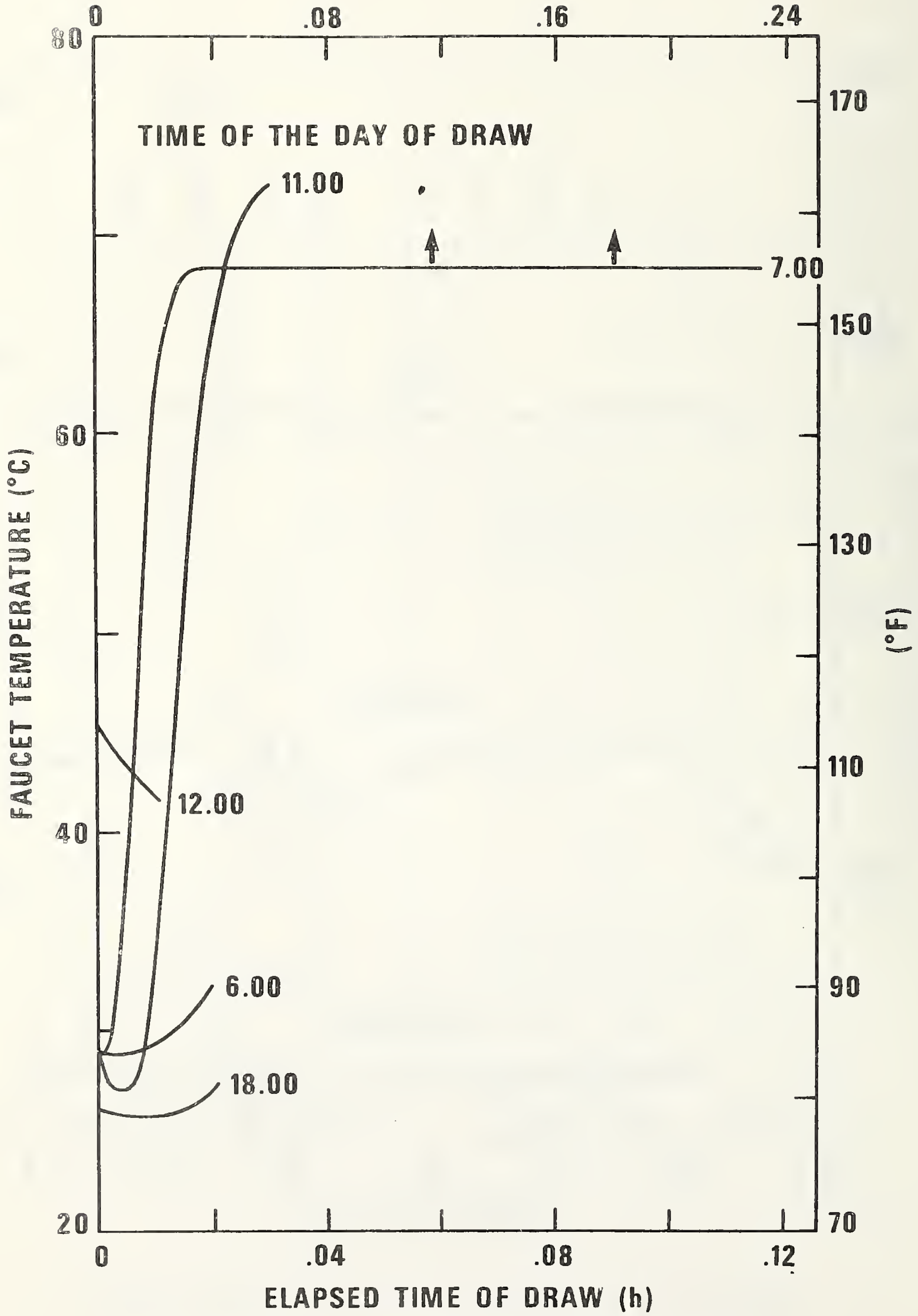


Figure 15 Faucet temperatures of draws, Case A.

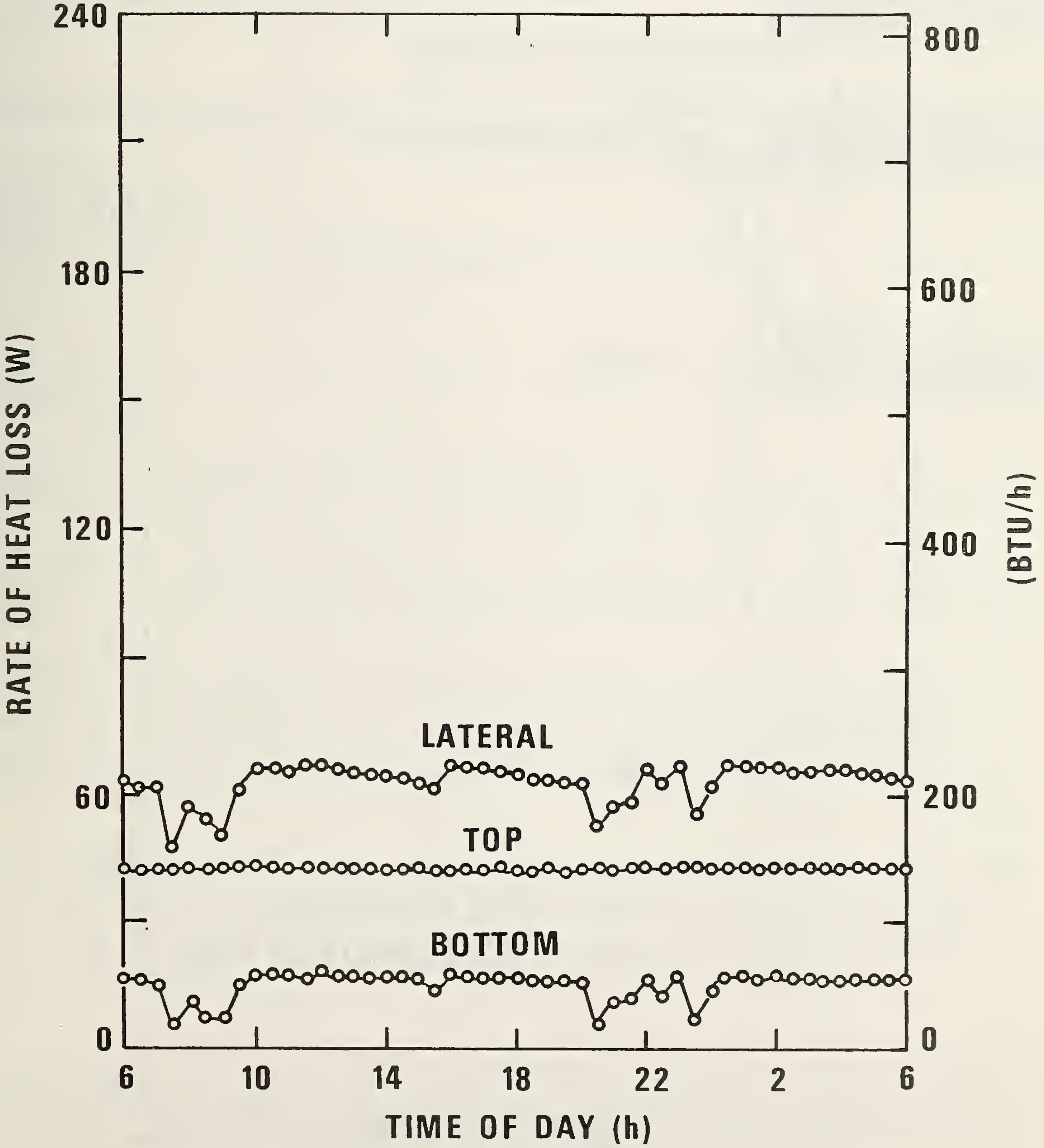


Figure 16 The rates of heat loss of water heater B.

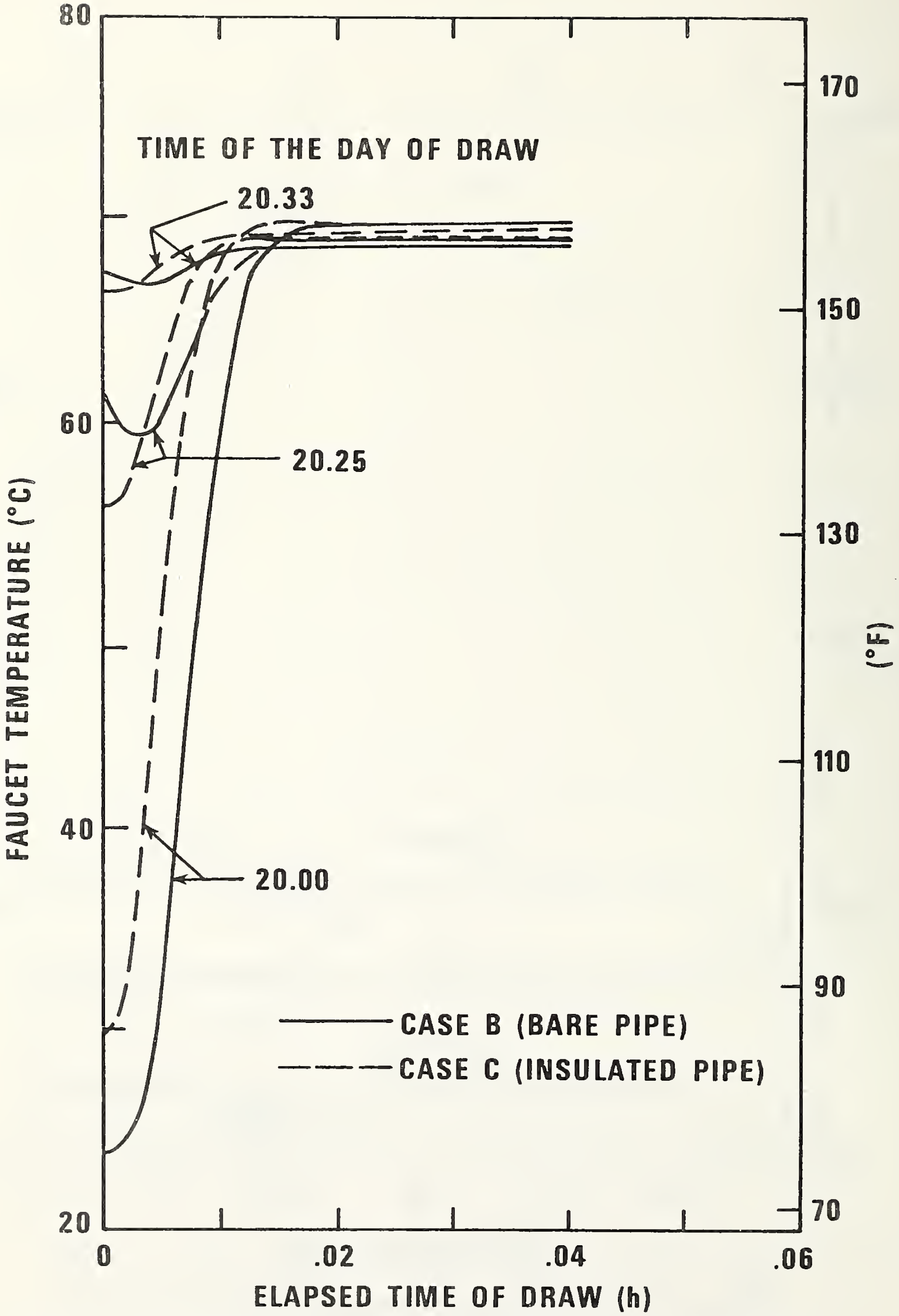


Figure 17 Faucet temperatures of dishwashing draws, Cases B and C. Elapsed time for piston flow of hot water to travel to the faucet = 0.00238 hour.

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15. SUPPLEMENTARY NOTES

☐ Document describes a computer program; SF-185, FIPS Software Summary, is attached.

16. ABSTRACT *(A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here.)*

A computer model, entitled NBS/ESDS, has been developed using a finite difference method to simulate the generation, storage and distribution of a hot water system commonly found in many residences, commercial facilities or industrial facilities.

The conduction heat equation which governs the heat transfer in the water heater during all modes of operation, and the heat transfer in the distribution pipe network during its cooldown periods, is solved. A solution is also obtained for the pipe flow equation which governs the heat transfer in the distribution pipe network during the draw periods. Thermal stratification of water is modeled. A plug flow model is used to simulate the draw of heated water from the water heater.

A parametric study is also conducted on a given hot water system for a given draw schedule during a 24-hour period. The effects studied include insulating the hot water heater, insulating the distribution pipe, and incorporating a heat trap. Effects of concentrating the distributed draws to two short periods of time of the day on the hot water delivery and the heat loss with the same energy consumption are also studied.

17. KEY WORDS *(six to twelve entries; alphabetical order; capitalize only the first letter of the first key word unless a proper name; separated by semicolons)*

Computer model, Distribution pipe, Dynamic response, Electricity, Energy, Explicit method, Faucet temperature, Finite difference, Heat loss, Implicit method, Insulation, Water heater.

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